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HEAT TRANSMISSION AND TUBE LENGTH IN MARINE FEED-WATER HEATERS.

BY LEO LOEB, M. E., ASSOCIATE.*

In the attempt to promote the economy and efficiency of all power machinery and to add to the scientific knowledge underlying the correct design of prime movers and auxiliaries, both stationary and marine, the general subject of heat transmission between two fluids separated by a metal wall has been critically investigated in recent years. The results of such investigations, both in the United States and abroad, have enabled designing engineers to grasp the principles underlying the distribution of heating surface in boilers to such an extent that stationary boilers operating on coal fuel and mechanically stoked show overall efficiencies as high as 76.2 per cent. when generating 7.4 pounds of equivalent steam per hour per square foot of heating surface. Naval practice has progressed to the point where an oil-fired water-tube boiler develops efficiencies of 79.0 and 76.5 per cent. when generating respectively 10.3

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and 18.7 pounds of equivalent steam per hour, discharging stack gases with little dilution and low carbon monoxide at stack temperatures of 561 and 759 degrees Fahrenheit.*

In the field of condenser practice the recent important contribution to scientific literature is " The Transmission of Heat in Surface Condensation," George A. Orrok, "American Society of Mechanical Engineers," 1910, p. 1139–1214.

Articles dealing with the economies effected by feed heating or the possibilities of increasing the economy of plants by the introduction of feed heaters are frequently seen. It is the intention in this instance, however, to develop the theory of heat transmission as applied to closed heaters operating on exhaust steam and to apply this theory with reliable data from tests of marine feed heaters to the preparation of design curves incorporating all the operating variables for a given type of heater. The results from practice naturally show less departure from the theoretical deductions than with other heat transfer apparatus, inasmuch as there is an absence of soot or scale deposits which require frequent interruptions in the case of boiler and evaporator researches, and freedom of the difficulties arising from presence of air in large volume at the pressures dealt with in experiments on condensers. Given the necessary mechanical equipment, tests may without great difficulty be conducted to determine effect of velocity, exhaust steam pressure or initial temperature as desired, since the problem resolves itself to a direct transfer of heat between condensing steam and water without, as a rule, the disturbing elements of scale or air.

I.—General Theory of Heat Transfer from Steam to Water.

When a metal tube carrying a moving column of water is exposed to the heating action of surrounding steam there occurs a physical state at each section which may be illustrated

*Oil Burning, J. J. Hyland, Lieutenant Commander, U. S. N., "Journal of American Society of Naval Engineers," May, 1914, p. 400.

by figure 1. The steam space has a constant temperature represented by the line AB. Immediately surrounding the tube is a film composed of particles of steam and water of condensation, and the resistance to heat flow therein results in a tem-

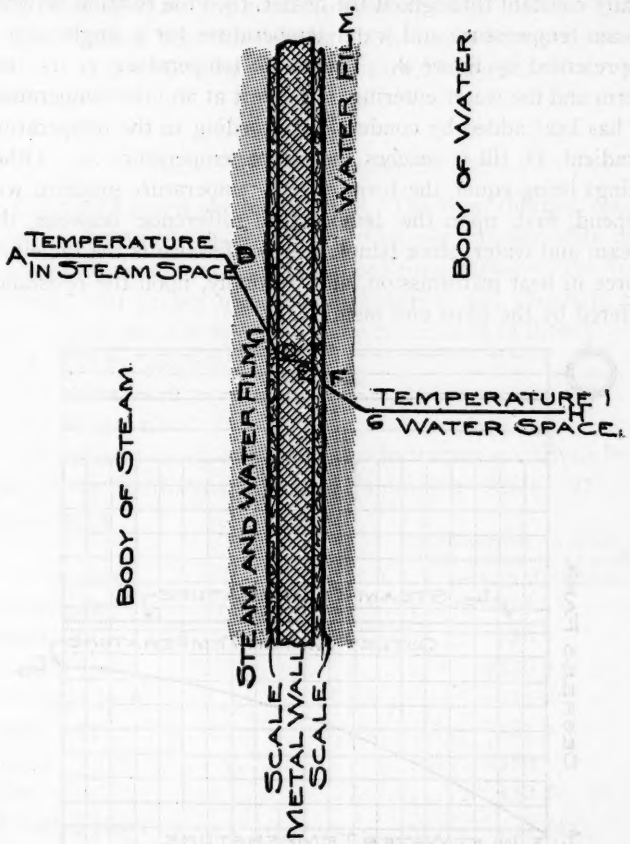


FIG. 1.

perature drop represented by the line BC; the tube wall may have a scale resulting from the process of manufacture or from the deposition of chemical salts which results in a temperature drop CD; the metal wall of the tube proper accounts for a slight temperature drop DE, while on the water side

there may again exist a thin scale and there will always be a water film with temperature drops EF and FG. The total drop in temperature is from AB in the steam space to GH in the body of the water. If the steam temperature be substantially constant throughout the heater, then the relation between steam temperature and water temperature for a single tube is represented by figure 2. The steam temperature, t_s is uniform and the water entering on the left at an inlet temperature t_i has heat added by conduction according to the temperature gradient, G, till it reaches the outlet temperature t_o. Other things being equal, the form of this temperature gradient will depend, first, upon the temperature difference between the steam and water, since temperature difference is the impelling force in heat transmission, and, secondly, upon the resistance offered by the films and metal walls.

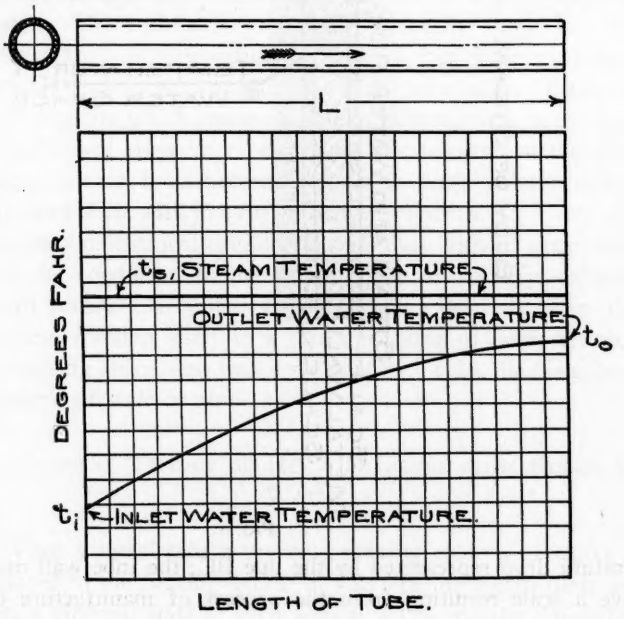


FIG. 2.

Heat resistance is best measured in terms of its reciprocal, K, thermal conductivity, the number of heat units transmitted

per unit area across a given space per unit of time. Denoting by

Q = heat transferred per hour in B.t.u.;
t_m = mean temperature difference;
S = area of heating surface in square feet;
K = rate of heat transmission in B.t.u. per hour per square foot per degree temperature difference;

then

$$Q = SKt_m.$$

II.—Thermal Resistances.

Considering first the nature of K: The total thermal resistance is dependent upon the two film resistances, scale and metal-wall resistances. Since proper preparation of the tube material and proper upkeep and operation will eliminate the scale conditions, the resistance may be reduced to three:

(a) Steam and water film;
(b) Metal wall;
(c) Water film.

The relative values of thermal conductivities as derived from data in the Smithsonian Physical Tables, 6th edition, 1914, are given below:

Material.	Conductivity in B.t.u. per hour per sq. ft. of surface per inch of thickness.
Aluminum	966.
Brass	736.6
Copper	2633.2
Wrought Iron	411.8
Steel, soft	321.9
Silver	3180.
Tin	412.7
Asbestos paper	1.247
Water	4.553
Salt solution	7.743
Air	.16476

Although this table does not include steam, the value for vapor will probably be above that of air but considerably below that of water. The values of thermal conductivity for

metals should be carefully noted. It may be observed that the transfer of heat through copper is 8.17 times as great as through steel of the same thickness, but since tube material is seldom over 1/10 inch thick, there will result 16,332 and 3,219 B.t.u. per hour per square foot per degree temperature difference for an average copper and an average steel tube. Since the usual coefficients of heat transmission will vary from 400 to 1,000 B.t.u. per hour per square foot of heating surface per degree average temperature difference between steam and water, it must be plain that the temperature difference on the two sides of the metal tube is very little; that tube material or tube thickness only slightly impedes heat flow, and that a water film .00173 inch thick will give the same resistance as a 1 inch thickness of copper tube. It is evident, therefore, that a metal tube will transmit all the heat that is presented to its surface and that the controlling resistances lie in the two films which cling to the metal surface. The condensing steam presents a wet surface, so that resistances on the two sides are much alike.

The formation of such a film is a friction effect. The microscopic irregularities of surface of the metal walls tear off particles of the fluids passing and prevent these particles from being swept along with the major current. The more completely these particles of water are dislodged and swept from the surface, the better able are other colder particles to replace them and absorb their share of heat from the surface. Hence the problem in producing high transmissions in heaters is the destruction of this water film by a scrubbing action produced by a high velocity along the heating surface. When the limit of heat transmission has been reached on the water side by a velocity that is entirely practical, the controlling resistance has passed to the steam side, and the only way to further increase heat transmission is to sweep away the film on the steam side.

III.—Temperature Differences.

A factor in the heat transfer equation of equal importance with the coefficient of conductivity is temperature difference.

This temperature difference can be increased in but one way, by raising the temperature of the heating medium, since the feed inlet is fixed in practice by the temperature of the condenser wet-pump discharge and hotwell drains. An increase in exhaust temperature is obtained by throttling the auxiliary exhaust, a method which off-hand would not seem to be conducive of best overall results. A further analysis of the situation may show this to be of some value. Direct-acting feed pumps as at present designed do not realize an economy from low back pressures, as apparently the restricted ports build up a certain minimum back pressure below which there is no reduction in steam consumption by a lower exhaust pressure. A careful analysis of this condition requires knowledge of steam consumption of pumps and blowers at varying speed and back pressures. Much test data on steam consumption of direct-acting pumps and turbine-driven blowers are on file at the Bureau of Steam Engineering, and curves of total heat consumption and boiler-heat economy could be drawn as a function of exhaust pressure. Their intersection would determine the best exhaust pressure for a given speed.

The type of temperature gradient resulting from the assumption that heat transfer is at any instant proportional to temperature difference is shown in the curve on figure 3.

In this figure let:

t = fluid temperature at any point;
l = length of tube measured from inlet;
W = weight of fluid per hour;
c = specific heat of fluid;
S = heating surface per unit of length of tube;
K = coefficient of heat transfer;
t_s = steam temperature;
t_i = inlet water temperature;
t_o = outlet water temperature;
t_m = mean temperature difference.

In a length of tube dl the fluid temperature will rise dt due to the temperature difference, $t_s - t$. The heat absorbed by the fluid in B.t.u. per hour will be

$$cW dt,$$

and the transfer across the metal wall due to temperature difference is

$$K(t_s - t)\,S dl.$$

Equating these two relations

$$cW dt = K(t_s - t)\,S dl,$$

$$\frac{dt}{t_s - t} = \frac{KS dl}{cW}.$$

Combining the constants K, S, c, W into another constant K′ and integrating:

$$\log_e (t_s - t) = \frac{KS}{cW} l + \text{constant},$$

which is a relation between heating surface and temperature variation in a heater tube if heat transmission is proportional to temperature difference. Between limits of inlet and outlet temperature t_i and t_o and for the entire length l', this becomes

$$\log_e \left(\frac{t_s - t_i}{t_s - t_o}\right) = K' l' = \frac{KS l'}{cW},$$

$$Q = KS l' t_m,$$

and

$$Q = cW(t_o - t_i).$$

Hence

$$K S l' t_m = cW(t_o - t_i);$$

But

$$\frac{KS l'}{cW} = \log_e \left(\frac{t_s - t_i}{t_s - t_o}\right).$$

Hence

$$t_m = \frac{t_o - t_i}{\log_e \left(\frac{t_s - t_i}{t_s - t_o}\right)}.$$

Using the above relation between temperatures and length of tube it is possible to construct the temperature gradient when the inlet and outlet feed temperatures are known.

For the full length of tube, l', and the temperature limits t_i and t_o ;

$$K'l' = \log_e\left(\frac{t_s - t_i}{t_s - t_o}\right).$$

For a fraction length, say $l = \frac{l'}{2}$.

$$K'\frac{l'}{2} = \log_e\left(\frac{t_s - t_i}{t_s - t_3}\right),$$

where t_3 is the temperature at half tube length.

$$2\log_e\left(\frac{t_s - t_i}{t_s - t_3}\right) = \log_e\left(\frac{t_s - t_i}{t_s - t_o}\right),$$

$$\log_e\left(\frac{t_s - t_i}{t_s - t_3}\right)^2 = \log_e\left(\frac{t_s - t_i}{t_s - t_o}\right).$$

Dropping the logs,

$$\left(\frac{t_s - t_i}{t_s - t_3}\right)^2 = \left(\frac{t_s - t_i}{t_s - t_o}\right),$$

$$t_s - t_3 = \sqrt{(t_s - t_i)(t_s - t_o)},$$

$$t_3 = t_s - \sqrt{(t_s - t_i)(t_s - t_o)}.$$

Similarly for other fractional lengths as $l = \frac{l'}{4}$, $t = t_4$,

$$(t_s - t_4) = \sqrt[4]{(t_s - t_i)^3(t_s - t_o)}.$$

In this way the temperature gradient for a given tube may be plotted, as the curve in figure 3, when inlet and outlet temperatures for a given steam pressure and water velocity are known, and this curve will determine the characteristics of shorter or longer heaters without knowing the numerical value of the coefficient of heat transfer.

The above analysis would be very satisfactory were it not for the fact that there exists considerable experimental evidence that heat transfer in feed heaters is not directly proportional to temperature difference but to some fractional power of the temperature difference. Tests on actual heaters where

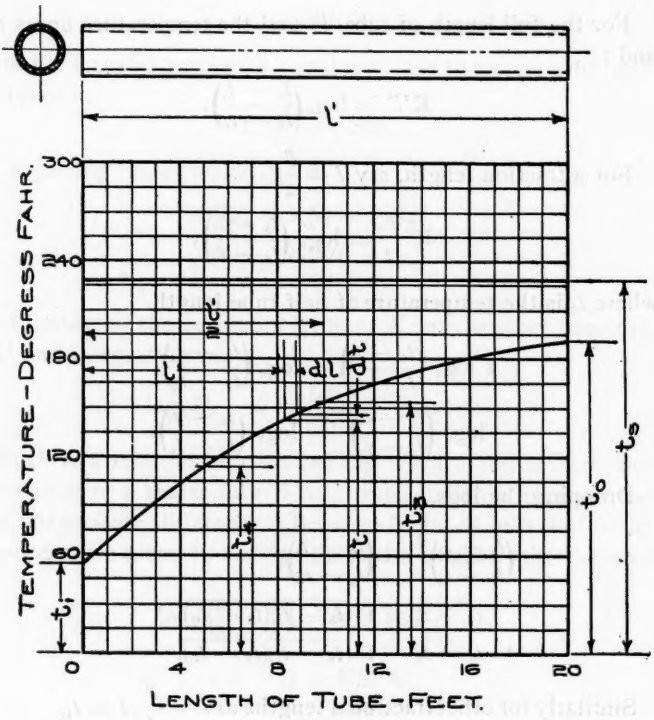


FIG. 3.

this fact appears will be discussed at a later time, but for the purpose of establishing the relation between temperature difference and heat transfer there are cited results obtained from a single-tube experimental heater tested under the direction of the writer at the Engineering Experiment Station. This heater consisted of a single ¾-inch copper tube. No. 18 B.W.G., four feet long, secured within a 4-inch pipe, which formed the steam space. The inlet could be regulated to any desired temperature by an auxiliary heater, and the temperature gradient at any given water rate was determined by noting inlet and outlet temperature at three-minute intervals for a period of 18 minutes. The inlet temperature was then raised

in the auxiliary heater to correspond closely with the average outlet temperature of the initial run and data taken as before.

Thus there could be determined the temperatures at four-foot intervals of a heater whose length could be varied to suit the temporary conditions of velocity and steam temperature. The data in Table I is from a typical set of tests with a steam pressure of 5 pounds gage and a water velocity of about four feet per second.

TABLE I.—TEMPERATURE GRADIENT IN A ¾-INCH COPPER TUBE.

Outside tube diameter, inch ..... .750
Inside tube diameter, inch ..... .655
Length, feet 4.0
Water-heating surface, square foot ..... .6853
Steam-heating surface, square foot ..... .7873

No.	Steam temperature, degrees F.	Water temperature, degrees F.		Average temperature difference.	Water heated per hour, pounds.	Heat transfer B.t.u. per hour per sq. foot steam surface.
		Inlet.	Outlet.			
1	2	3	4	5	6	7
1	228.4	55.6	93.4	143.9	2,070	99,060
2	228.0	93.2	125.2	118.8	2,095	85,039
3	228.4	125.1	152.3	89.7	2,106	72,771
4	228.1	152.5	173.4	65.1	2,120	56,007
5	227.3	174.4	189.8	45.2	2,106	41,453

The gradient corresponding to columns 2, 3 and 4 is plotted in figure 4. Although the temperature rise per pass varies from 15.4 to 37.8, the average arithmetical temperature difference may be very closely considered to be the mean temperature difference per pass, since the curvature of the temperature gradient is very slight and a straight line connecting any two points varies only slightly from the best smooth curve through the points.

The relation between temperature difference and heat transfer per hour, columns 5 and 7 of Table I, is plotted on logarith-

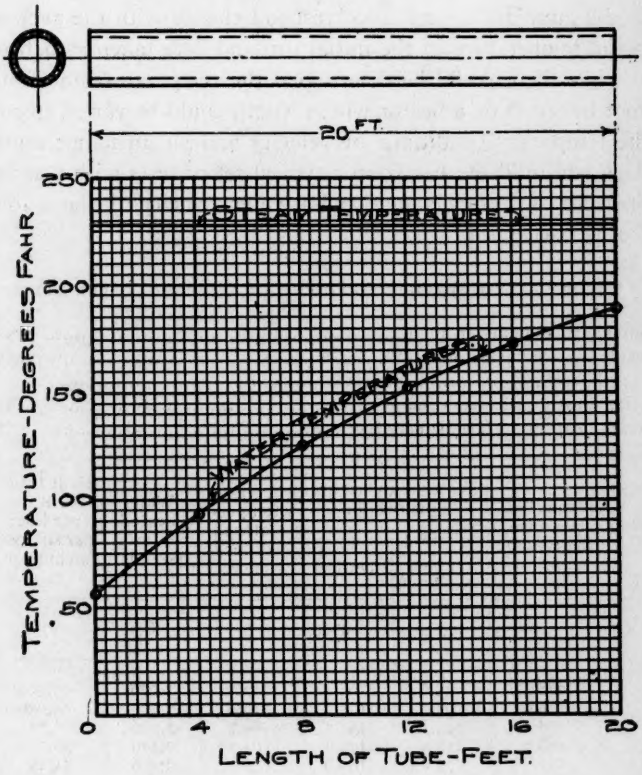


FIG. 4.

mic cross section paper in figure 5. Allowing for slight variations in velocity these points fall quite accurately on a straight line, which is the equivalent of saying that heat transfer in feed-water heaters is proportional to some power of the instantaneous temperature difference. Using the same symbols as before, the result may be expressed analytically as:

$$Q = KS(t_s - t)^n,$$

where n is the slope of the line on the logarithmic curve, in this case a value somewhat less than 1. Referring to figure 3 and

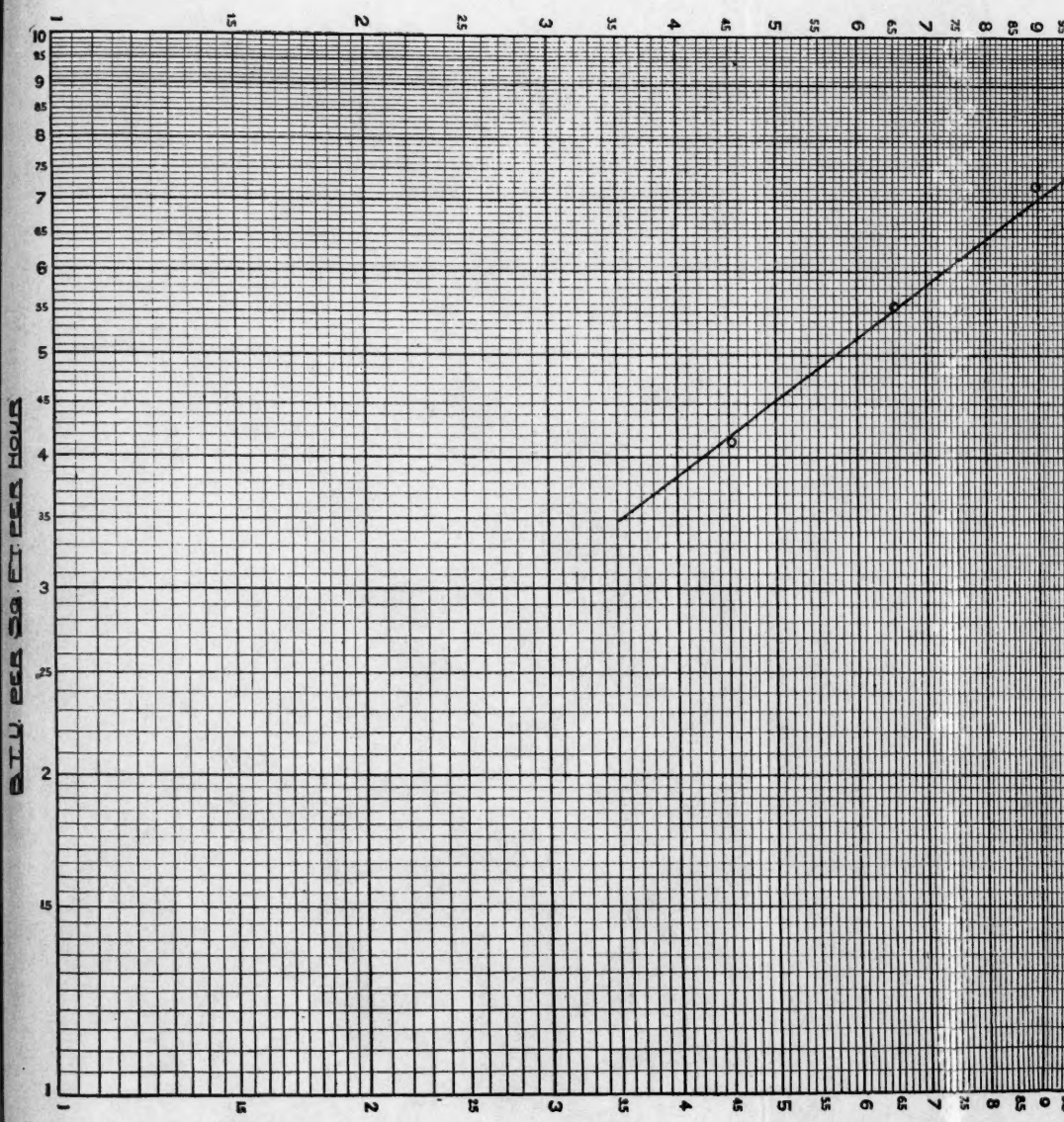


FIG.5

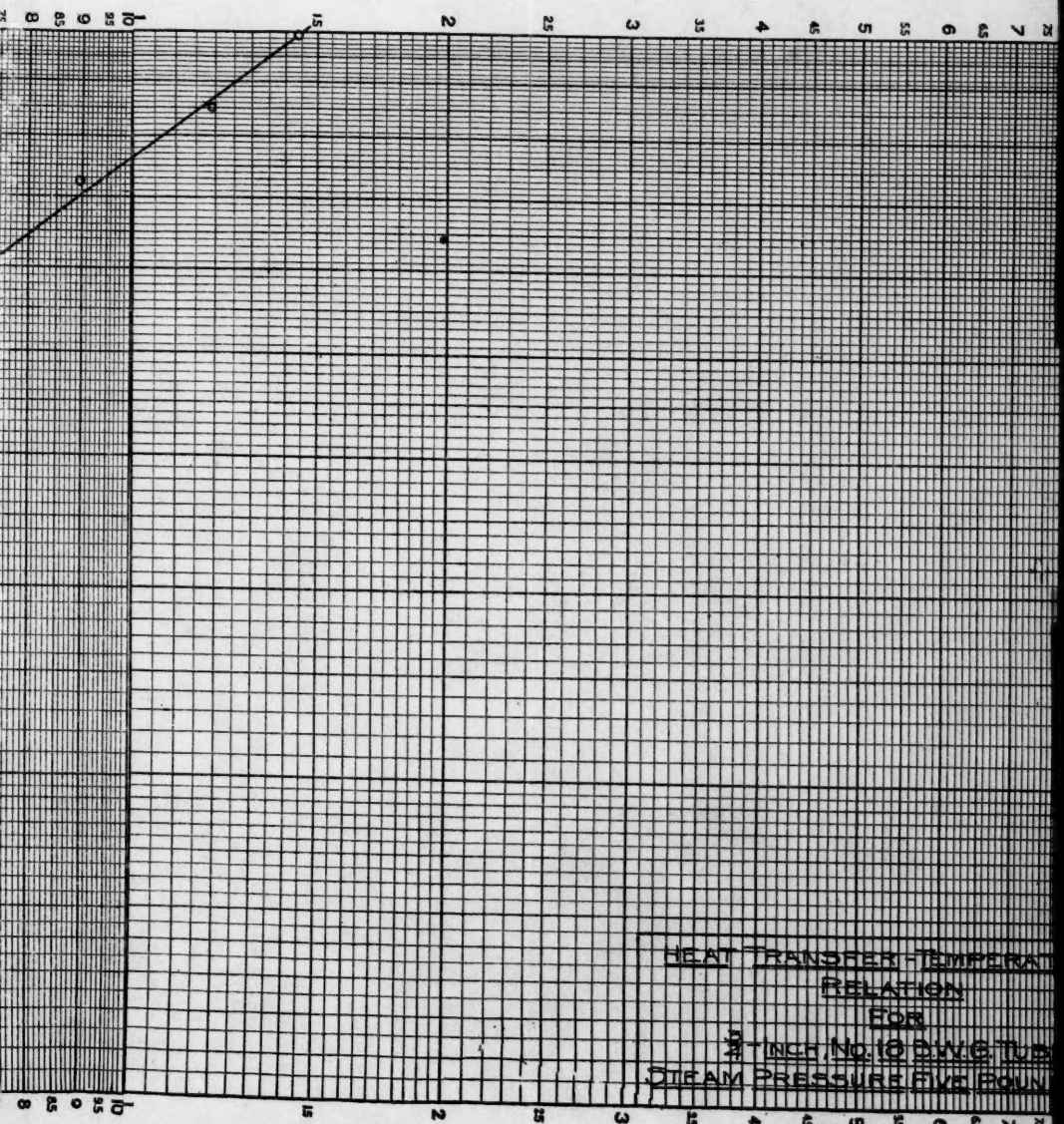
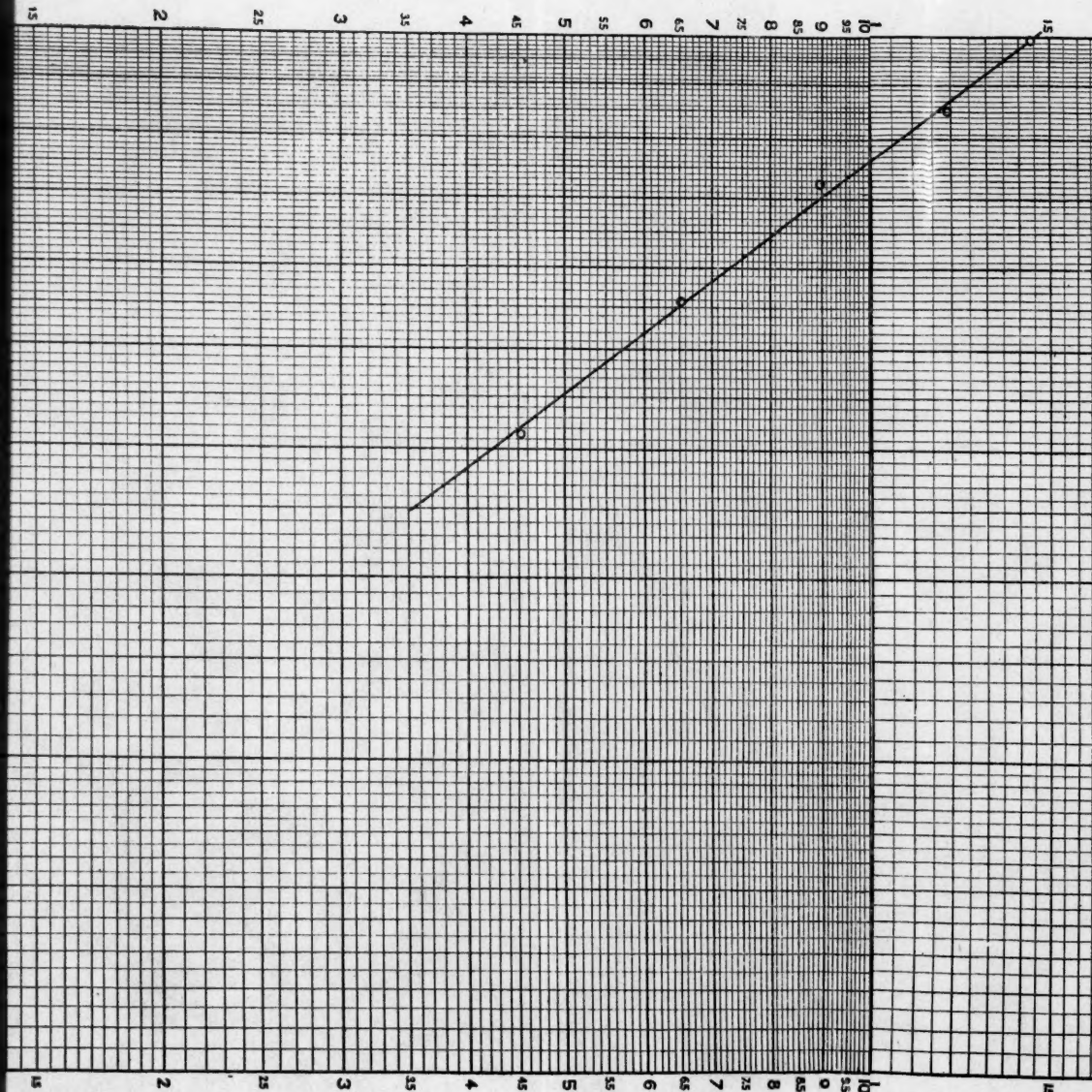

HEAT TRANSFER - TEMPERA
RELATION
FOR
INCH No. 10 B.W.G. TUB
STEAM PRESSURE FIVE POUN
DIFFERENCE - DEGREES FAHR.

FIG. 5.



TEMPERATURE DIFFERENCE - DEGREES

FIG. 5.

HEAT TRANSFER - TEMPERATURE RELATION FOR ¾ INCH No. 18 B.W.G. TUBE. STEAM PRESSURE FIVE POUNDS GAGE.

EGREES FAHR.

proceeding as in the previous case, the heat added in an element of length is:

$$dQ = cW dt = KS(t_s - t)^n \, dl,$$

$$\frac{dt}{(t_s - t)^n} = \frac{KS dl}{cW}.$$

Integrating:

$$\frac{1}{n-1}(t_s - t)^{1-n} = \frac{KS}{cW} l + \text{constant}.$$

Substituting between limits of t_i and t_o for a length of tube l',

$$\frac{1}{1-n}\left[(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}\right] = \frac{KS}{cW} l',$$

$$Q = KSl'(t_m)^n = cW(t_o - t_i),$$

$$\frac{KSl'}{cW} = \frac{t_o - t_i}{(t_m)^n} = \frac{1}{1-n}\left[(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}\right].$$

Hence,

$$t_m = \frac{(1-n)(t_o - t_i)}{(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}}.$$

This formula is the same as that derived by Mr. Orrok for condensers, but is obtained from the basic experimental proof that heat transfer is proportional to a power of the temperature difference instead of the secondary fact that rate of heat transmission per degree temperature difference is proportional to a power of temperature difference. The latter method is somewhat more involved, inasmuch as it introduces another variable factor, U, which varies with temperature. The relation of U as given in the second column of Table I, page 1152 of Mr. Orrok's paper* may be omitted, as it is without significance in the theory. The important thing is the value of K, column 1 of Mr. Orrok's table, and this is tacitly admitted in the curves of temperature rise, page 1153, where there are presented curves of variable K and N.

*A. S. M. E., 1910.

Furthermore, there is a decided disadvantage in obtaining U as function of t, because the whole purpose is to obtain an experimental value of K *which will remain constant throughout* the heater design in question.

The value of U in Mr. Orrok's equation is the coefficient which would have to be inserted in order that the correct basic form,

$$Q = K(t_s - t)^n S,$$

may be converted to and have the same numerical value as

$$Q = U(t_s - t)\, S.$$

Hence

$$\frac{U}{K} = (t_s - t)^{1-n},$$

$$U = K(t_s - t)^{1-n}.$$

This makes U, the coefficient of heat transmission, a function of the temperature difference and introduces an undesirable complication, because the natural thing to do is to search for a coefficient which will remain constant for a given film agitation. Naturally for fractional values of n the value of K in formula on page 267 will be higher than the value derived from the logarithmic mean, the amount of variation being greater for departures of the exponent, n, from unity.

A relation between the coefficients, K and K′, from the two equations may be established as follows:

$$\frac{KSl'}{cW} = \frac{1}{1-n}\left[(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}\right],$$

$$\frac{K'Sl'}{cW} = \log_e \frac{(t_s - t_i)}{(t_s - t_o)},$$

$$\frac{K}{K'} = \frac{(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}}{(1-n)\log_e \dfrac{(t_s - t_i)}{(t_s - t_o)}},$$

$$= \frac{(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}}{\log_e \left(\dfrac{t_s - t_i}{t_s - t_o}\right)^{1-n}}.$$

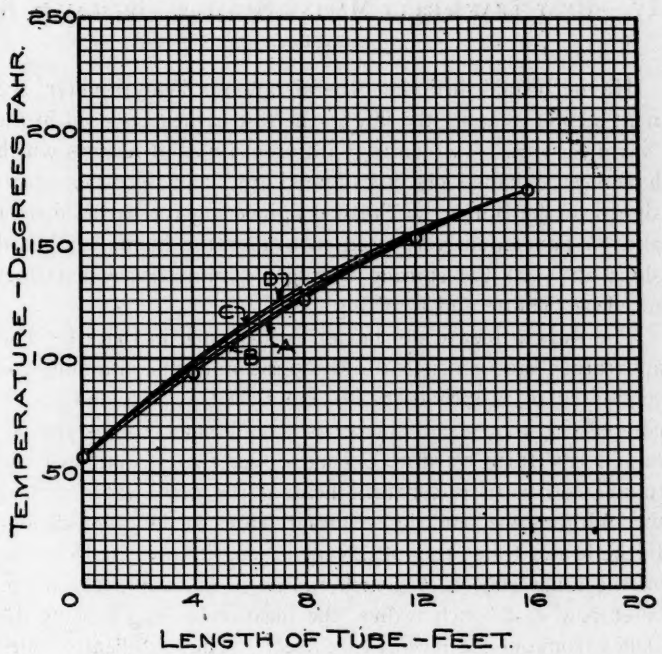


FIG. 6.

A comparison of the two laws of temperature variation for the range of data in Table I is shown in the curves of figure 6. The lines A, B and C are the gradients from the relation

$$l = c(t_s - t)^n$$

when the values of n are 0.7, 0.8 and 0.9. The line D is the gradient from the relation

$$l = C \log_e (t_s - t)$$

and the small circles mark the inlet and outlet points in columns 2 and 3 of Table I.

IV.—Heat Transfer in Marine Heaters. Bureau S. E. Heater.

Having established the general law for heat transfer, the next step is to apply the law to the test of heaters used in the Naval Service. Two entirely different types of heaters which have been exhaustively investigated are (1) a feed heater designed by the Bureau of Steam Engineering for battleships 34 and 35 and extensively used in evaporator plants on board ship; and (2) the spirally corrugated film heater manufactured by Schutte and Koerting.

The Bureau feed-water heater is illustrated in Plate I. The apparatus tested consists of a composition shell containing 117 semi-circular ¾-inch tubes No. 16 B.W.G., expanded into a composition tube sheet, and a cast-steel bonnet, the bonnet being cast to form separate passages over the two ends of the tubes. When used for feed heating the feed water passes through the tubes and the steam circulates in the outer casing. In the heater tested the tubes vary in length from 20.64 inches on the inner row of 5⅜-inch radius to 69.72 inches on the outer row of 21-inch radius, the mean tube length being 45 inches from outside face of tube sheet. The total heating surface is 88.2 square feet, of which 86 square feet is tube area. The net area through the water passage of tubes is .362 square foot.

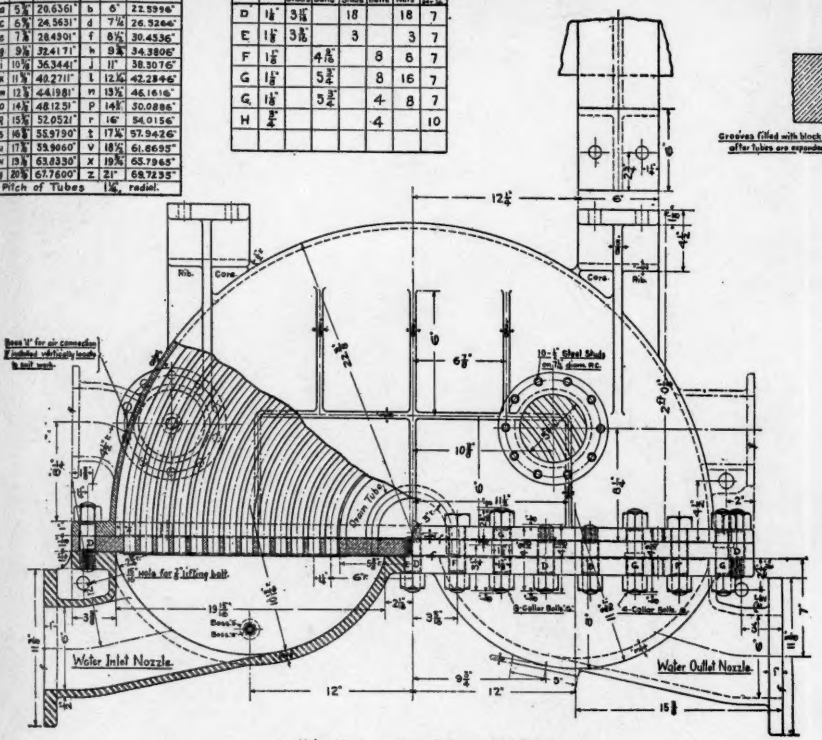
The original tests of this heater, conducted by Mr. O. Z. Howard, M. E., were reported in the Journal of the American Society of Naval Engineers, February, 1912, pp. 155–166.

The method of test and the test apparatus is described in detail in the reference above and there is reprinted here, as Table II, the data and results of forty one-hour tests. The tests are divided into two main groups dependent on the inlet temperature, which was maintained at about 80 degrees F. in the first case and from 130 to 150 degrees in the second. Further subdivision was according to feed velocity and steam pressure. The mean water velocities chosen were 35, 71, 107,

V
2
1
2
M
A
Y
1
5
XUM

Table of Tube Lengths					
Letter	Radius	Length	Letter	Radius	Length.
a	5 3/8"	20.6361"	b	6"	22.5996"
c	6 5/8"	24.5631"	d	7 1/16"	26.5266"
e	7 3/8"	28.4901"	f	8 1/16"	30.4536"
g	9 1/8"	32.4171"	h	9 3/8"	34.3806"
i	10 5/8"	36.3441"	j	11"	38.3076"
k	11 3/8"	40.2711"	l	12 1/16"	42.2846"
m	12 3/8"	44.1981"	n	13 1/16"	46.1616"
o	14 1/8"	48.1251"	p	14 3/8"	50.0886"
q	15 3/8"	52.0521"	r	16"	54.0156"
s	16 3/8"	55.9791"	t	17 1/16"	57.9426"
u	17 3/8"	59.9061"	v	18 1/2"	61.8695"
w	19 3/8"	63.8331"	x	19 3/4"	65.7965"
y	20 3/8"	67.7601"	z	21"	69.7235"
Pitch of Tubes 1 1/16" radial.					

Table of Bolts and nuts.						
Mark	Size	Length		Number		
		Studs	Bolts	Studs	Bolts	Nuts	Washers
D	1 1/8"	3 5/16"		18		18	7
E	1 1/8"	3 5/16"		3		3	7
F	1 1/8"		4 7/16"		8	8	7
G	1 1/8"		5 1/4"		8	16	7
G	1 1/8"		5 1/4"		4	8	7
H	3/4"				4		10



Note: Heater as drawn for horizontal installation.
Nozzles shown in dash and double dot for vertical installation.

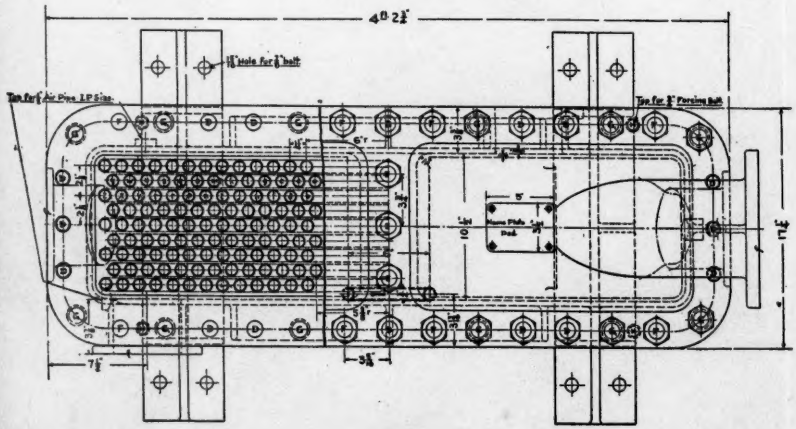
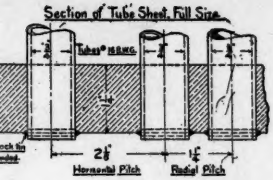
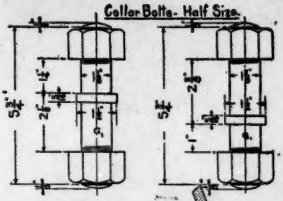


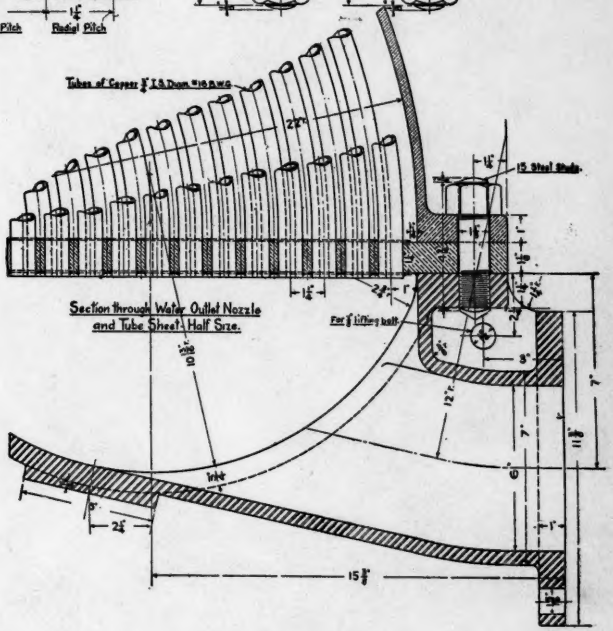
PLATE I.

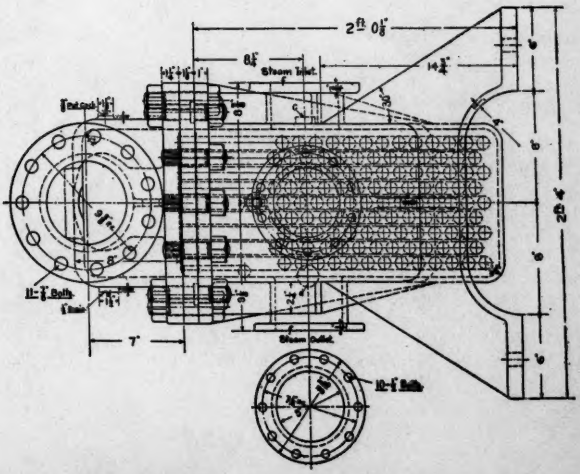
Section of Tube Sheet. Full Size.



Collar Bolts. Half Size.




Tubes of Copper ¾" I.D. Down # 18 B.W.G.
27"
15 Steel Studs.
Section through Water Outlet Nozzle and Tube Sheet. Half Size.
For ¾" lifting bolt
Inlet
15⅜"



143 and 171 feet per minute, corresponding to deliveries of about 48,000, 96,000, 143,000, 190,000 and 230,000 pounds of water per hour or 410, 820, 1,220, 1,622 and 1,960 pounds of water per tube per hour. The steam pressures were 5, 10, 15 and 20 pounds gage and the water pressure was held at from 230 to 250 pounds gage. The tests show in general an increase of heat transfer with steam pressure and with velocity, but a decrease at the same pressure with increasing temperature difference. These tests do not form as good material for analysis as those which followed; furthermore, some doubt is cast due to irregularities in data caused, as was subsequently determined, from the accumulation of air in the steam space.

Table III gives the data from additional tests which were made at a constant water velocity of 130 feet per minute, constant steam pressure of 10 pounds gage and variable temperature of inlet water. The first test was run with an inlet temperature of 75 degrees, the outlet temperature from this test being used as the inlet temperature of the second test, and so on, with the inlet of each succeeding test corresponding closely to the outlet of the previous run until an outlet temperature of 208 degrees F. was reached.

The tubes were then fitted with retarders consisting of annealed copper strips, 5/8 inch wide and 0.0268 inch thick, twisted into a spiral of six inches pitch, after which the series of tests was repeated. The drop in pressure through each pass of the heater as measured by differential mercury gage was 0.55 inch of mercury without retarders as against 0.98 with retarders, hence the introduction of the retarders involved no serious increase in friction. Subsequently the tests were extended under the writer's direction to water rates of 2,025, 2,500 and 3,030 pounds of water per tube. The inlet water temperature was taken in the water inlet nozzle and the outlet at two points in a lagged discharge pipe far enough beyond the outlet nozzle to insure the thorough mixing of the water by a spiral deflector or mixer. The water was measured by a 6-inch × 3-inch Venturi meter, which indicated according to

Number of test.	Date.	Duration of test (hours).	Barometer, corrected.	Room temperature, degrees F.	Water side.							Steam side.	
					River water pumped through heater, pounds per hour.	Equivalent fresh water pumped through heater, pounds per hour.	Temperature at inlet, degrees F.	Temperature at outlet, degrees F.	Degrees rise.	B.T.U.s absorbed by water, per hour.	Pressure at pump, pounds gage.	Steam condensed, pounds per hour.	Pressure at inlet, inches mercury.
1	2	3	4	5	6	7	8	9	10	11	12	13	14
	May.												
1	6	1	29.66	79.0	49248	47574	84.2	121.8	37.6	1788782	281.7	1917.5	9.08
2	6	1	29.66	79.0	48721	47064	86.3	124.7	38.4	1807258	226.6	1957.5	19.59
3	6	1	29.66	78.0	49690	48001	88.7	128.9	40.2	1929640	236.0	2100.0	29.58
4	6	1	29.59	78.0	49630	47943	90.8	136.7	45.9	2200584	228.9	2416.0	38.34
5	8	1	29.44	71.0	99580	96194	82.8	112.9	30.1	2895439	241.0	3054.5	9.06
6	8	1	29.39	71.0	99550	96165	82.7	113.8	31.1	2990732	238.4	3185.0	19.24
7	8	1	29.30	75.0	99003	95637	83.9	119.3	35.4	3385550	245.3	3624.5	29.08
8	8	1	29.33	73.0	98943	95579	87.1	126.3	39.2	3746697	249.6	4029.0	38.60
9	9	1	29.17	76.0	149625	144538	77.1	97.6	20.5	2963029	236.6	3121.0	9.41
10	9	1	29.14	76.5	149550	144465	77.9	101.7	23.8	3438267	239.3	3635.5	19.06
11	9	1	29.10	76.0	149943	144845	81.1	107.7	26.6	3852877	244.3	4108.0	29.34
12	9	1	29.08	75.0	148872	143810	81.1	114.3	33.2	4774492	258.6	5101.0	38.70
13	10	1	29.24	83.0	200357	193545	79.1	102.2	23.1	4470890	251.9	4605.0	9.00
14	10	1	29.22	83.0	199380	192601	77.2	103.2	26.0	5007626	254.0	5211.0	19.08
15	10	1	29.22	83.0	198583	191831	78.7	108.2	29.5	5659015	257.0	5935.0	29.27
16	11	1	29.31	85.5	199160	192389	79.9	112.3	32.4	6233404	252.6	6590.0	38.30
17	11	1	29.30	85.0	239352	231214	79.7	100.2	20.5	4739887	245.7	4932.0	9.43
21	24	1	29.47	75.5	239420	231280	79.1	103.0	23.9	5527592	234.3	5781.0	19.05
22	24	1	29.46	77.0	239462	231320	80.2	107.4	27.2	6291904	225.7	6602.0	29.24
23	24	1	29.41	73.0	238944	230820	81.7	112.2	30.5	7040010	222.1	7434.0	39.27
18	12	1	29.38	85.0	49385	47706	147.2	171.4	24.2	1154485	249.1	1260.0	8.86
19	12	1	29.37	85.0	49390	47711	144.3	170.2	25.9	1235715	259.1	1362.0	19.09
20	12	1	29.31	85.0	49350	47672	143.7	174.1	30.4	1449229	248.9	1599.0	29.28
	June.												
36	1	1	29.09	82.5	49328	47651	151.8	185.0	33.2	1582013	234.1	1756.0	39.25
	May.												
28	29	1	29.46	86.0	99969	96571	139.2	156.2	17.0	1641707	244.1	1769.0	10.48
29	29	1	29.45	87.0	99495	96112	134.7	154.9	20.2	1941462	241.1	2097.0	20.00
30	29	1	29.44	87.0	99277	95902	133.3	155.6	22.3	2138615	255.0	2329.0	30.71
31	29	1	29.44	87.0	98930	95566	135.6	160.4	24.8	2370037	254.3	2592.0	39.45
32	31	1	29.36	82.0	147376	142365	145.6	161.1	15.5	2206658	253.9	2330.0	9.39
33	31	1	29.36	80.0	146867	141874	144.3	162.5	18.2	2582107	246.0	2754.0	19.48
34	31	1	29.23	81.0	146763	141773	146.4	166.2	19.8	2807105	240.4	3015.0	29.05
35	31	1	29.26	81.0	147631	142612	148.4	170.6	22.2	3165986	247.9	3415.0	39.37
	June.												
37	1	1	29.12	84.0	195842	189183	149.8	163.1	13.3	2516134	249.4	2652.0	9.61
38	1	1	29.12	81.5	197820	191094	147.8	161.6	13.8	2637097	252.4	2814.0	19.50
39	1	1	29.15	81.5	195743	189088	148.1	163.6	15.5	2930864	237.6	3123.0	29.66
40	1	1	29.15	81.5	195604	188953	150.4	168.6	18.2	3438945	242.0	3690.0	38.89
	May.												
24	25	1	29.35	78.5	237447	229374	128.3	132.3	4.0	917496	244.4	990.0	9.89
25	25	1	29.35	78.5	237632	229553	131.0	144.8	13.8	3167831	248.4	3362.0	19.90
26	25	1	29.31	79.8	238008	229916	132.2	148.1	15.9	3655664	250.7	3899.0	30.00
27	25	1	29.30	81.0	237546	229460	129.8	147.0	17.2	3946712	258.7	4213.0	38.86

Steam side.													
Pressure at inlet, pounds absolute.	Temperature at inlet, degrees F.	Temperature at inlet corresponding to pressure, degrees F.	Degrees superheat.	Temperature of outlet steam water to weighing tanks, degrees F.	B.T.U.s rejected by steam.	Radiation from heater, B.T.U.s per hour.	Radiation from heater, calculated from test, B.T.U.s per hour.	Equivalent dry saturated steam condensed, pounds per hour.	Velocity of water through tubes, feet per minute.	Heat transfer per sq. ft. heat'g surf. per deg. differ. (temp. due to pressure, column 17).	Mean temperature difference, geometrical (temperature due to pressure, column 17).	Heat transfer per sq. ft. heat'g surf. per deg. mean temp. diff. (Temp. due to pressure.)	
15	16	17	18	19	20	21	22	23	24	25	26	27	
19.03	227.1	226.0	1.1	218.4	1858878	70096	71088	1934	35.3	148.6	122.3	165.8	
24.20	239.3	238.3	1.0	230.1	1882880	75622	76944	1975	35.0	140.4	131.2	156.1	*a*
29.09	249.2	248.6	0.6	239.6	2006747	77107	82176	2121	35.7	142.3	137.3	159.3	
33.38	257.6	256.8	0.8	251.7	2285758	85174	86208	2430	35.7	156.1	142.5	175.0	
18.96	232.5	224.9	7.6	216.9	2974313	78874	77520	3091	71.4	237.2	125.9	260.7	
23.89	240.9	237.5	3.4	227.7	3073945	83213	81552	3223	71.4	225.1	139.4	243.2	*b*
28.67	250.9	247.7	3.2	238.6	3470705	85155	84432	3665	71.0	240.2	143.4	267.6	
33.20	262.8	256.2	6.6	248.3	3836551	89854	91104	4077	71.1	257.2	149.4	284.3	
18.94	231.9	225.0	6.9	214.8	3044573	81544	74832	3165	107.1	233.3	138.1	243.2	
23.66	241.3	237.0	4.3	226.2	3515398	77131	79104	3685	107.1	250.4	143.8	271.0	*c*
28.70	250.9	247.8	3.1	238.3	3934720	81843	83952	4156	107.4	267.5	152.9	285.6	
33.28	259.5	256.3	3.2	245.2	4864660	90168	88560	5170	106.7	314.7	160.4	337.4	
18.78	249.3	224.6	24.7	211.7	4544932	74042	79824	4723	143.5	354.1	132.8	381.6	
23.72	258.1	237.2	20.9	224.0	5092356	84730	84048	5338	142.8	360.7	149.4	379.9	*d*
28.72	253.9	247.8	6.1	231.2	5735519	76504	82032	6058	142.3	384.4	154.8	414.4	
33.21	264.5	256.2	8.3	242.8	6316957	83553	85920	6712	142.8	406.1	156.5	451.5	
19.02	235.3	225.3	10.0	212.9	4828428	88541	72144	5020	171.3	375.9	138.1	389.1	
23.83	239.8	237.4	2.4	224.0	5598043	70451	78864	5869	171.3	400.8	144.4	433.9	*e*
28.83	258.3	248.1	10.2	233.3	6380160	88256	87024	6740	171.6	420.7	156.3	456.3	
33.63	269.6	256.9	12.7	242.8	7143502	103492	94368	7595	171.3	462.1	160.0	498.6	
18.78	225.4	224.6	0.8	218.9	1220038	65553	67392	1268	36.0	178.7	65.1	201.0	
23.80	238.1	236.3	1.8	229.6	1310467	74752	73488	1373	36.0	161.5	78.7	178.0	
28.77	248.9	247.9	1.0	241.1	1525590	76361	78672	1611	36.0	165.9	88.5	185.6	*f*
33.56	258.4	256.8	1.6	250.3	1664484	82471	84432	1770	36.1	179.7	87.7	204.5	
19.62	229.9	226.9	3.0	218.9	1716355	74648	69072	1786	72.6	221.8	79.0	235.6	
24.29	242.2	238.5	3.7	232.4	2015167	73705	74496	2114	72.2	220.0	93.9	234.4	
29.54	253.3	249.4	2.9	243.7	2219120	80505	79344	2347	72.0	216.6	103.7	233.8	
33.84	258.4	257.3	1.1	251.5	2453688	83651	82272	2609	71.8	228.5	107.3	250.4	
19.04	238.1	225.3	12.8	216.6	2275581	68923	74928	2366	107.2	323.6	72.1	346.9	
23.99	247.9	237.8	10.1	228.3	2665453	83346	80592	2795	106.9	323.1	84.6	346.0	
29.62	250.0	249.6	0.4	238.8	2884137	77032	81120	3050	106.9	316.8	92.1	345.5	
33.71	259.9	257.1	2.8	248.2	3246545	80559	85872	3452	107.6	338.5	96.1	373.4	
19.02	230.5	225.3	5.2	216.9	2579590	63456	70320	2682	142.6	387.3	69.8	408.6	
23.88	240.1	237.5	2.6	225.0	2722680	85583	76128	2855	144.0	344.0	83.4	358.4	
28.89	262.0	248.2	13.8	233.9	3021696	90832	86640	3192	142.5	342.2	93.7	354.6	
33.47	257.5	256.6	0.9	243.5	3521518	82573	84480	3743	142.6	375.8	95.5	408.2	
19.28	226.5	226.0	0.5	180.3	997168	79672	71040	1037	171.6	115.7	102.0	102.0	
24.19	239.3	238.2	1.1	223.9	3254846	87015	77184	3414	172.0	344.1	98.7	363.8	
29.13	250.1	248.6	1.5	235.1	3745126	89462	81744	3958	172.4	364.6	107.1	386.9	
33.47	257.8	256.6	1.2	241.8	4028420	81708	84864	4282	171.9	360.1	123.0	363.7	

Temperature difference over 122 degrees.

Temperature difference below 102 degrees.

TABLE III.—TEST OF

(With Velocity of 131 feet per minute,

Test number.	Date, September, 1911.	Duration of test, hours.	Barometer, corrected.	Room temperature, degrees F.	Water side.							Steam side.			
					River water pumped through heater, pounds per hour.	Equivalent fresh water pumped through heater, pounds per hour.	Temperature at inlet, degrees F.	Temperature at outlet, degrees F.	Degrees rise.	B.T.U.s absorbed by water, per hour.	Pressure at pump, pounds gage.	Steam condensed, pounds per hour.	Pressure at inlet, inches mercury.	Pressure at inlet, pounds absolute.	Temperature at inlet, degrees F.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
															WITH
1	20	1	29.97	76.0	178543	172473	75.0	108.2	33.2	5726104	55	5943	19.46	24.28	252.6
2	20	1	29.96	79.0	179933	173815	108.9	135.6	26.7	4640861	50	4870	19.48	24.29	252.2
3	21	1	30.13	74.5	176311	170316	136.3	158.8	22.5	3832110	50	3978	20.10	24.67	267.5
4	21	1	30.11	75.0	178260	172199	158.8	176.5	17.7	3047922	50	3208	19.52	24.37	255.2
5	21	1	30.09	75.0	178357	172293	175.6	189.6	14.0	2412102	50	2665	19.70	24.46	244.5
6	21	1	30.07	77.0	175999	170015	189.5	200.8	11.3	1921170	50	2278	19.80	24.50	262.6
7	22	1	30.09	78.5	175731	169756	200.1	208.1	8.0	1358048	50	1802	19.95	24.45	243.4
															WITHOUT
8	26	1	30.03	81.0	178041	171988	78.2	105.6	27.4	4712471	50	4997	19.29	24.23	246.7
9	26	1	30.03	80.0	175948	169966	104.3	128.8	24.5	4164167	55	4523	19.23	24.20	244.4
10	27	1	30.17	77.0	178356	172292	130.4	149.9	19.5	3359694	50	3744	19.47	24.37	243.9
11	27	1	30.16	71.0	176922	170907	149 9	166.9	17.0	2905419	53	3237	19.41	24.34	243.6
12	27	1	30.08	73 0	175459	169493	167.9	182.3	14.4	2440699	50	2753	19.42	24.31	242.0
13	27	1	30.07	74.0	173111	167225	182.7	194.1	11.4	1906365	55	2321	19 30	24.24	242.2
14	28	1	30.11	74.0	171906	166061	192.8	200.5	7.7	1278670	55	1748	19.20	24.21	245.2
15	28	1	30.10	75.0	171695	165857	200.5	206.8	6.3	1044899	50	1552	19.36	24.29	240.5

FEED-WATER HEATER.

and steam pressure of 25 pounds absolute.)

Steam side.												
Temperature at inlet, corresponding to pressure, degrees F.	Degrees superheat.	Temperature of outlet steam water to weighing tank, degrees F.	B.t.u.s rejected by steam.	Equivalent dry saturated steam condensed, pounds per hour.	B.t.u.s unaccounted for.	Component velocity of water through tubes (parallel to axis) feet per minute.	Heat transfer per hour per sq. ft. of heating surface, per degree difference (temp. due to press., col. 17).	Mean temperature difference, geometrical (temp. due to press., col. 17).	Heat transfer per hour per sq. ft. heating surface, per degree mean temperature difference (temp. due to press., col. 17).	Pounds of dry steam condensed per sq. ft. of heating surface.	27 reduced to condensation per sq. ft. of tube surface (86 square feet).	Temperature at inlet, degrees F.
17	18	19	20	21	22	23	24	25	26	27	28	16
RETARDERS.												
238.44	14.16	231.7	5744979	6028	18875	135.7	397.1	146.19	443.99	68.33	69.33	252.6
238.47	13.73	232.0	4705297	4937	64436	137.4	406.0	116.34	452.17	55.96	56.78	252.2
239.34	28.16	232.4	3870674	4063	38564	135.4	421.6	91.43	475.10	46.06	46.75	267.5
238.65	16.55	231.8	3104702	3258	56780	137.7	432.7	70.80	487.98	36.93	37.49	255.2
238.86	5.64	232.1	2564653	2691	152561	138.4	432.2	56.00	488.24	30.50	30.94	244.5
238.95	23.65	233.6	2208589	2318	287419	136.9	431.7	43.58	499.70	26.27	26.56	262.6
239.13	4.27	232.5	1729902	1816	371854	137.4	394.4	34.98	440.07	20.58	20.88	243.4
RETARDERS.												
238.33	8.37	224.4	4852641	5091	140170	130.5	333.6	146.29	365.14	57.71	...	246.7
238.26	6.14	229.9	4362660	4577	198493	130.0	355.0	121.70	387.86	51.88	...	244.4
238.65	5.25	232.1	3602028	3780	242334	131.8	351.8	98.09	388.24	42.85	...	243.9
238.58	5.02	232.6	3112181	3265	206762	131.4	371.4	79.92	412.08	37.01	...	243.6
338.51	3.49	233.8	2641448	2771	200749	131.0	391.8	63.19	437.82	31.41	...	242.0
238.35	3.85	233.9	2226976	2336	320611	129.8	388.3	49.67	435.06	26.48	...	242.2
238.28	6.92	333.0	1681279	1764	402609	129.2	318.7	41.49	349.34	19.99	...	245.2
238.47	2.03	233.3	1488772	1562	443873	129.4	311.9	34.71	341.23	17.71	...	240.5

calibrations before and after tests 1.5 per cent. low at all rates.

Steam was delivered from a low-pressure steam line to the 5-inch steam supply where its temperature and pressure were taken. There was always some superheat, hence a calorimeter was not needed. Steam pressure was again taken at the bottom of the steam space. Air cocks were fitted at both top and bottom of the steam space and allowed to blow continuously. Only in this way could consistent results be obtained, as considerable air accumulated in the lower portion of the steam space and reduced heating effect if allowed to remain. In a subsequent section the effect of this air in steam will be treated in greater detail.

The results of the second series of tests are given in Table IV for plain tubes and in Table V for tubes with retarders. It should be noted that in several tests the steam pressure at inlet, column 15, is higher than the ten pounds fixed on for the test. This was necessary because at low water temperatures and high rates of flow the steam pressure drop from top to bottom of heater, as shown by a comparison of columns 15 and 17, was so great that with 10 pounds on the shell the outlet pressure was not sufficient to deliver the condensate through the cooling coil to the weighing tanks and maintain a water-free condition throughout the heating surface. However, in almost all of these cases the average steam temperatures, column 22, corresponding to the two pressures was practically 238 degrees, the saturation temperature of 10 pounds gage. The pressure drop within the steam space is further indicated by column 24, temperature of condensed steam, which is far below the temperature of saturation, column 20, corresponding to inlet pressure and even lower than that of column 22, corresponding to outlet pressure.

The data from these tests have been plotted on logarithmic cross-section paper in figure 7, where the abscissae are temperature differences in degrees F., and the ordinates are heat transfer in B.t.u. per hour per square foot of heating surface.

There can be no doubt that for the heater in question the heat transfer—temperature relation is given by:

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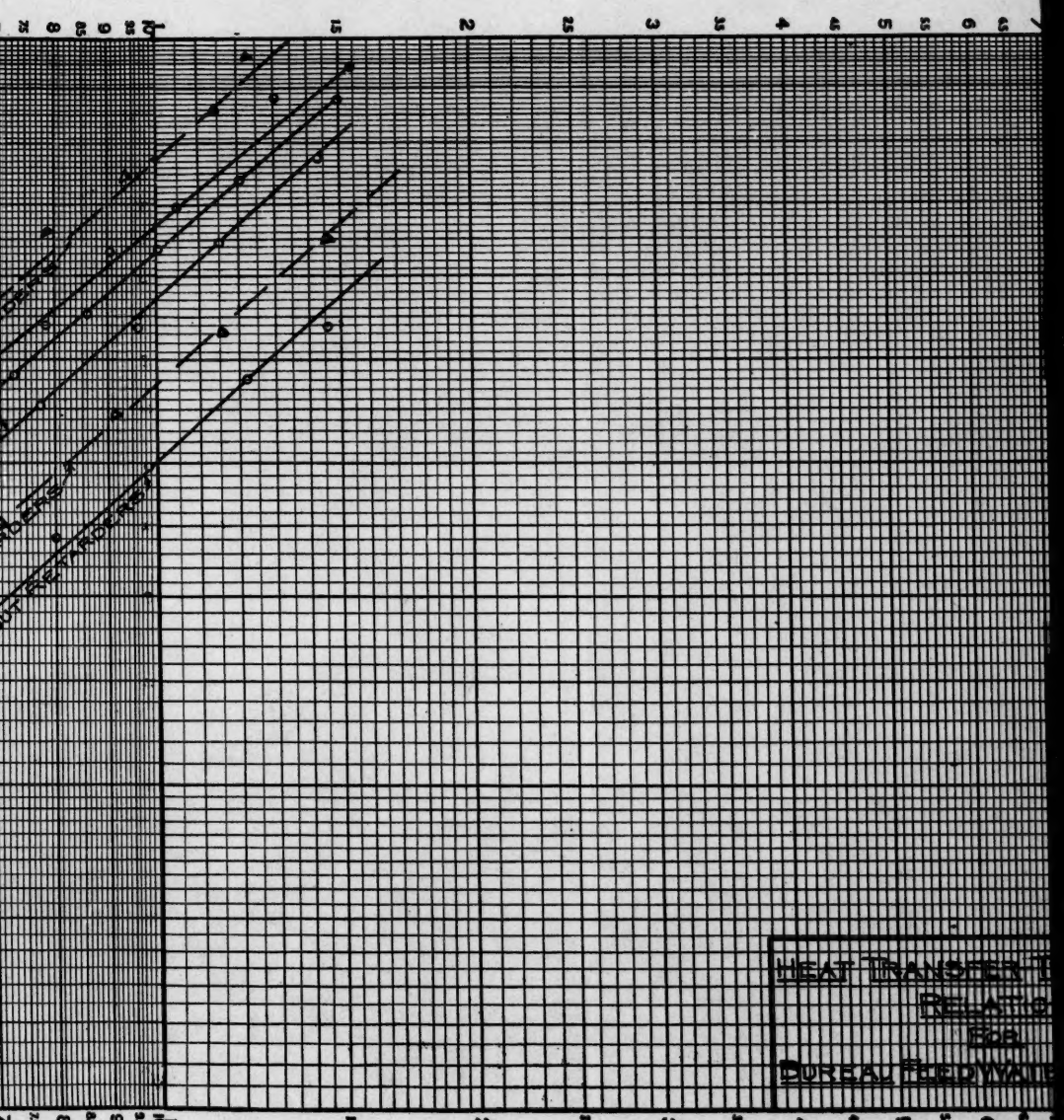
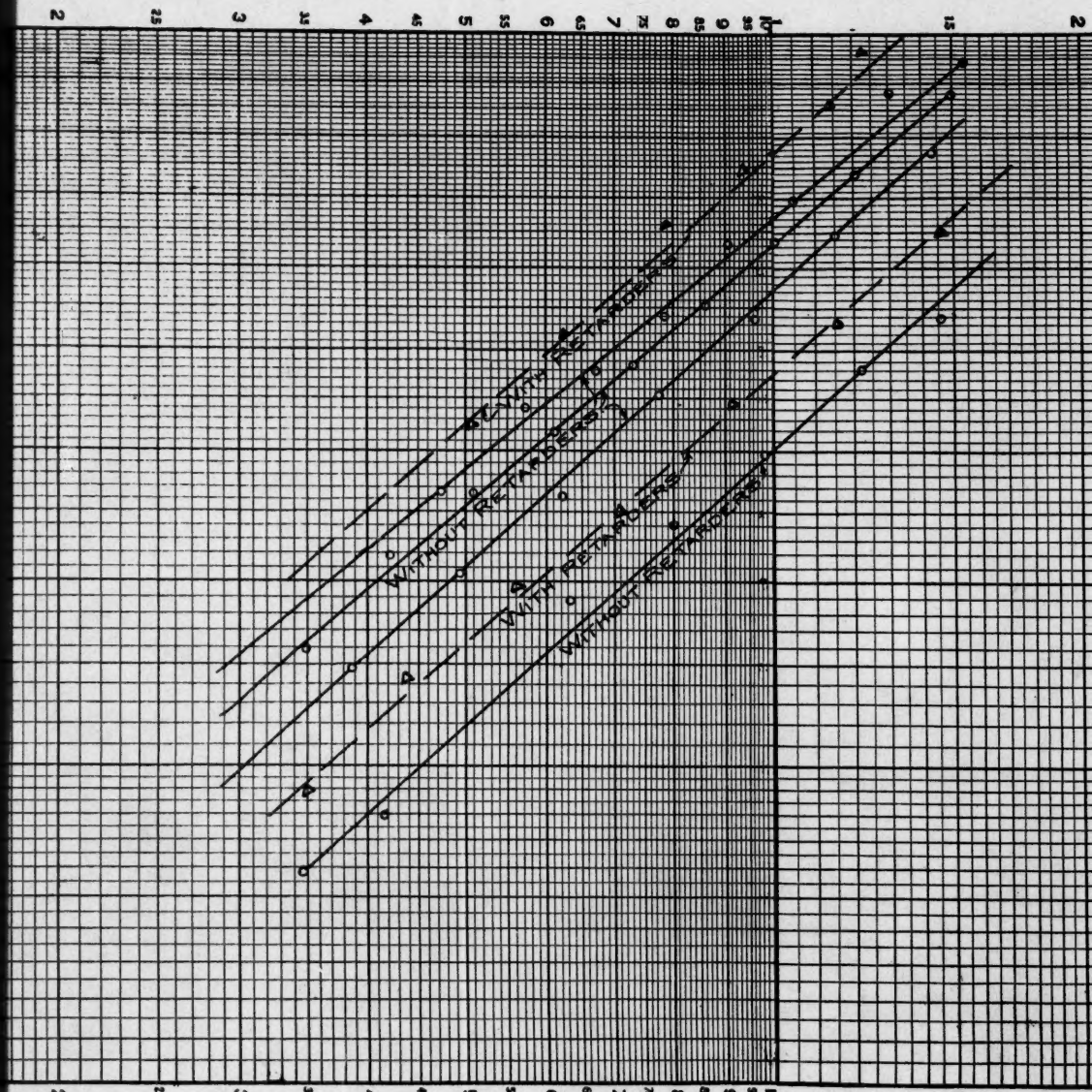


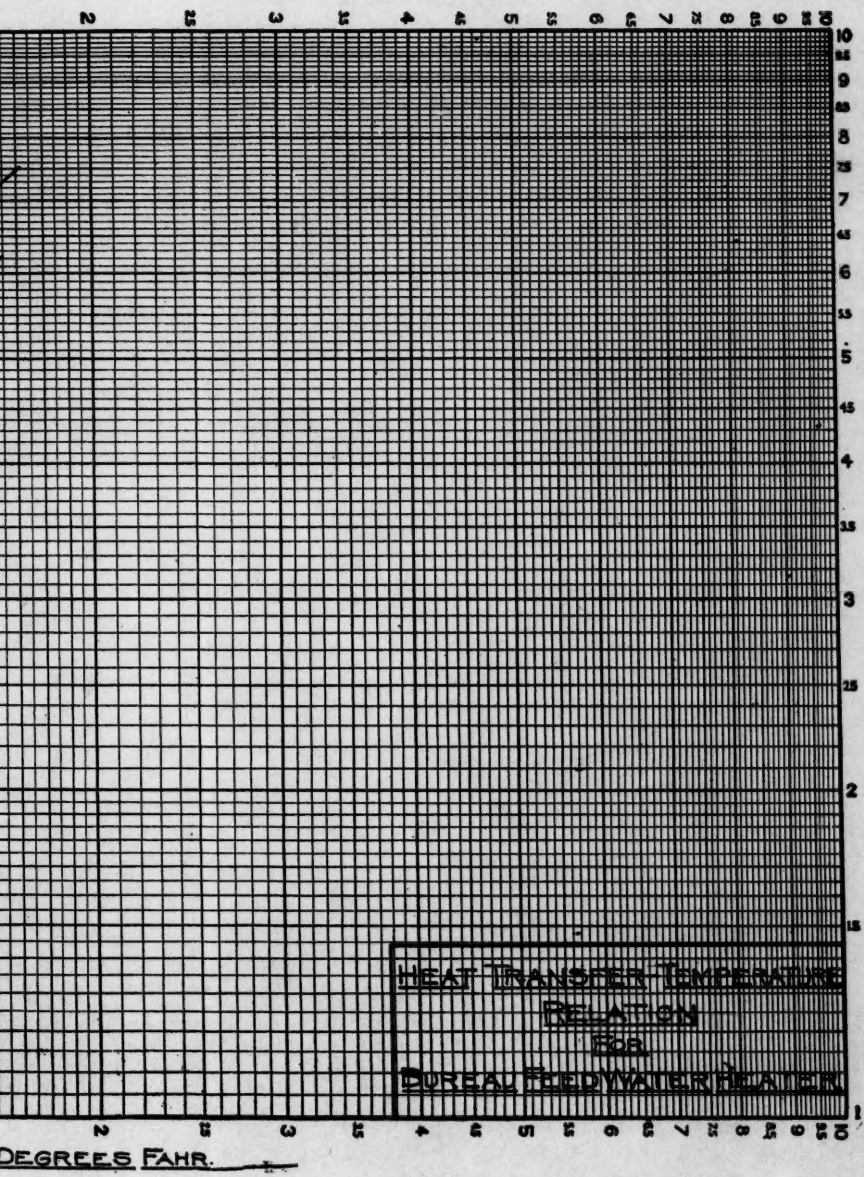
PERATURE DIFFERENCE - DEGREES FAHR.

FIG. 7.



TEMPERATURE DIFFERENCE - DEGREE

FIG. 7.



DEGREES FAHR.

Number of Test	Date, June Per., 1915.	Duration of Test. Minutes.	Barometer. Inches of Mercury.	Room Temperature.	Water Data.							Steam Data.														Natural Data.							
					Water Rate per Hour. Pounds per Hour.	Temperature at Inlet, °F.	Temperature at Outlet, °F.	Temperature Rise	Specific Heat of Water at Mean Temp.	B.t.u. Absorbed by Water per Hour.	Water Temp. Rise, Inches Hg.	Friction Loss, Inches of Mercury.	Friction Loss, Pounds per Square Inch.	Pressure at Inlet, Pounds per Square Inch Gage.	Pressure at Inlet, Pounds per Square Inch Absolute.	Pressure at Outlet, Pounds per Square Inch Gage.	Pressure at Outlet, Pounds per Square Inch Absolute.	Temperature, Corresponding to Inlet Pressure, °F.	Temperature, Corresponding to Outlet Pressure, °F.	Degrees Superheat.	Quality of Steam Corresponding to Inlet Pressure.	Average Steam Temperature, °F.	Temperature of Condensed Steam, °F.	Steam Condensed per Hour, Pounds.	Steam Condensed per Hour, Pounds.	Heat Rejected per Pound of Steam, B.t.u.	B.t.u. Rejected by Steam per Hour.	Radiation per Square, B.t.u. per Hour.	Equivalent Dry and Saturated Steam, Pounds per Hour.	Actual Velocity, Feet per Minute.	Mass, Pounds per Hour.	B.t.u. Transmitted per Square Foot per Hour.	Average Air Heating Temperature Difference.	B.t.u. per Square Foot per Hour per Degree Mean Temperature Difference.	Per Cent per Square Foot.	B.t.u. per Square Foot per Hour per Degree Mean Temperature Difference.	Per Cent per Square Foot.
1	2	3	4	5	6	7	8	9	10.	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38
1	31	30	29.93	62	236879	75.3	105.1	29.8	.968	6833939	9.5	0.39	0.18	10.76	25.47	4.36	19.07	213.6	218.1	32.5	205.4	233.3	228.1	3396.0	6796	990.0	6726040	-109319	6664	175	2025	77463	143.1	148.0	542.2	542.1	85.1
2	31	30	29.90	64	236913	106.3	131.4	25.1	.956	5607758	9.5	0.40	0.18	9.76	24.45	5.63	18.32	216.7	221.7	24.9	196.8	233.8	222.5	2937.5	5875	981.2	5764550	66732	5731	176	2025	64565	114.9	114.6	562.1	563.6	84.2
3	31	30	29.88	66	237400	129.2	150.2	21.0	.951	4746876	9.7	0.45	0.20	9.89	24.57	7.39	17.18	217.1	221.1	18.6	203.3	236.2	226.6	2683.0	4942	974.1	4814008	73124	4852	177	2023	53738	96.5	96.2	556.9	556.6	81.0
4	31	30	29.81	69	238830	150.5	168.3	17.8	.945	4009552	9.6	0.40	0.18	9.99	24.64	8.77	15.92	219.3	223.3	16.1	215.5	238.4	230.0	2145.0	4290	967.6	4151004	141452	4380	179	2037	45449	78.0	77.6	582.7	585.7	18.1
5	31	30	29.75	71	237614	168.2	186.2	14.3	.940	3393532	9.6	0.40	0.18	9.88	24.50	8.84	15.66	220.8	224.8	13.7	217.8	239.7	234.5	1714.5	3429	959.0	3288411	94873	3432	179	2032	36398	62.4	62.1	580.1	582.9	34.4
6	31	30	29.74	71	237240	186.8	204.8	12.0	.937	2847934	9.0	0.40	0.18	9.03	24.60	9.38	15.22	221.3	224.2	8.1	237.8	238.5	235.3	1434.5	2869	957.6	2747354	52480	2872	181	2044	30467	49.7	49.4	613.0	616.7	12.0
7	31	30	29.71	72	238452	204.6	224.6	9.8	.933	2779451	7.9	0.46	0.21	9.96	24.56	9.48	14.28	223.6	226.1	0.5	238.4	238.0	236.4	1180.0	2360	955.6	2255216	77265	2360	181	2038	24705	39.1	38.8	631.8	636.7	9.9
8	1	30	29.95	59	236309	75.0	102.0	27.0	.968	7746517	11.2	0.56	0.25	13.60	28.32	5.81	20.93	220.0	227.0	33.0	209.4	236.2	216.8	3368.5	7737	993.8	7687731	142274	8048	219	2533	67795	140.7	143.5	586.5	587.3	27.2
9	1	30	29.99	59	236090	102.7	128.6	22.9	.962	6536941	11.2	0.87	0.40	11.03	25.77	4.96	19.70	218.6	221.7	36.9	227.2	234.5	220.9	3368.0	6736	989.6	6705530	108509	6876	219	2531	57869	120.3	120.2	614.0	614.6	23.2
10	1	30	30.04	58	237305	128.6	143.0	18.2	1.010	5677984	10.9	1.14	0.52	10.11	24.87	5.72	19.46	268.2	239.8	86.4	227.2	234.5	229.0	2739.0	5998	980.7	5882839	854855	6070	221	2541	63793	100.6	100.3	634.1	636.0	20.4
11	1	30	30.04	60	236646	142.4	156.3	15.9	1.038	4867961	10.6	1.12	-0.52	10.08	24.84	7.19	21.65	265.4	239.7	25.7	233.0	236.4	229.1	2672.5	5345	975.3	5211973	345018	5404	222	2535	55179	86.0	85.6	642.6	644.6	18.2
12	1	30	30.06	61	237676	157.4	171.4	14.0	1.026	4868683	10.7	1.20	0.55	10.15	24.91	8.24	23.00	262.7	239.6	22.8	235.5	237.4	230.0	2736.5	5473	971.1	4737750	269067	4715	223	2541	48398	73.3	73.2	660.1	661.0	15.9
13	1	30	30.11	60	236917	170.9	183.1	12.2	1.022	3699356	10.3	1.14	0.52	10.13	24.92	8.81	23.60	261.9	239.6	22.0	236.9	238.4	234.5	2340.0	4011	968.2	3883450	183864	4047	223	2538	43955	61.4	61.1	689.0	686.3	13.6
14	1	30	30.15	60	237390	182.0	192.6	10.6	1.020	3834786	11.1	1.21	0.55	9.94	24.75	8.92	23.83	259.1	239.5	23.6	237.2	238.4	235.3	1733.5	3467	963.2	3339414	124618	3658	224	2542	36460	51.1	51.0	713.1	714.5	11.7
15	1	30	30.19	60	236555	198.1	201.4	3.3	1.016	2656206	11.2	1.26	0.57	9.98	24.81	9.39	24.22	244.6	239.7	4.9	238.3	239.0	236.5	1446.5	2897	957.8	2774247	-31439	2903	224	2538	31808	42.2	42.3	735.7	735.5	9.8
16	2	30	30.05	54	236864	73.5	97.9	24.4	1.041	6330048	13.1	1.76	0.81	15.06	29.81	6.39	21.14	250.4	249.9	40.5	231.0	240.5	217.9	4230.5	8461	997.3	8438155	116087	8605	263	3050	94331	155.8	153.6	613.3	614.1	24.1
17	2	30	29.96	56	236733	98.1	119.1	21.0	1.037	5770080	13.1	1.80	0.82	13.71	28.44	6.11	20.84	268.4	247.3	41.1	230.2	238.8	221.0	3824.0	7648	993.5	7596588	86908	7981	264	3049	85074	130.8	130.2	676.5	677.0	22.4
18	2	30	29.91	58	236456	118.3	135.9	16.6	1.032	5157413	13.5	1.83	0.83	9.98	24.68	4.39	19.09	260.0	239.4	40.6	225.2	238.3	219.7	3378.0	6756	991.8	6703581	146468	6464	267	3072	69734	104.7	104.6	666.6	667.8	18.0
19	2	30	29.85	60	237954	136.1	152.2	15.1	1.029	5564695	13.7	1.76	0.80	10.01	24.67	5.63	20.25	274.7	239.3	35.4	228.8	234.1	224.6	2901.5	5803	984.3	5711893	149268	5830	267	3075	63052	70.4	70.4	697.5	697.5	16.5
20	2	30	29.81	61	238976	156.9	163.8	12.9	1.027	4756432	13.5	1.80	0.82	10.02	24.67	7.23	21.88	268.2	239.3	28.9	232.8	236.1	227.9	2543.5	5087	977.8	4974269	219637	5148	269	3068	53914	76.7	76.8	685.1	684.8	14.3
21	2	30	29.76	62	238688	164.0	175.5	11.5	1.022	4211734	13.6	1.87	0.85	9.96	24.59	8.04	22.67	256.8	239.2	17.6	234.7	237.0	228.6	2207.0	4414	970.7	4286670	174836	4445	269	3064	47745	67.2	67.3	710.6	711.6	12.4
22	2	30	29.76	63	238695	175.4	186.0	10.6	1.021	3808811	13.4	1.87	0.90	10.15	24.77	8.71	23.33	252.1	239.6	13.5	236.3	238.0	231.0	2049.0	4098	967.3	3963995	81184	4131	270	3069	44012	57.9	58.1	728.2	729.8	11.5
23	2	30	29.72	65	239696	186.3	195.1	8.8	1.020	3523507	11.6	1.90	0.87	10.04	24.64	9.18	23.78	250.2	239.3	10.9	237.3	238.2	234.5	1716.5	3433	962.6	3304606	73093	3447	271	3073	36629	47.6	47.4	769.6	772.8	9.6
24	4	30	30.04	59	359696	194.9	201.9	7.0	1.018	2545330	10.8	1.73	0.79	10.08	24.78	9.25	24.62	246.2	239.6	6.6	239.2	238.4	235.0	1338.0	2676	960.1	2569208	23918	2684	269	3051	28691	41.0	40.8	703.8	707.1	7.5
25	4	30	30.04	59	362069	201.0	208.0	7.0	.930	2605336	11.2	1.69	0.77	10.01	24.77	9.88	24.64	243.7	239.6	4.1	239.3	239.5	236.8	1287.0	2454	957.1	2348783	63387	2498	265	3000	27904	35.0	34.9	740.2	742.3	7.0

U. S. NAVAL ACADEMY.

TABLE IV.

TEST OF BURKAU HEATER WITH RETARDERS.

TABLE V.

TEST OF BUREAU HEATER WITHOUT RETARDERS.

$$Q = K(t_s - t)^n$$

where

Q = B.t.u. per hour per square foot of H. S.;

K = a constant from experiment;

t_s = average saturation steam temperature within the heater;

t = water temperature;

n = a constant from experiment.

Furthermore n is the slope of the curve on the logarithmic plot and has for this heater a value

$$n = 0.85,$$

which is independent of the velocity and of the use of retarders. The value of K, which is the intersection of the line prolonged to the vertical axis when $(t_s - t) = 1$, is dependent on the velocity and is therefore the function which varies with the more or less efficient removal of the water side film. As illustrated here the use of retarders has the same effect as increasing the velocity by increasing the value of K. The few inconsistencies in points of the second curve are due to the fact that the heater stood for two months between the tests in Table IV and those in Table V and the tubes were probably slightly coated on the water side. It required at least two runs to thoroughly clean the tubes. Before the tests of Table IV the tubes were scoured, cleaned with a wire brush and hot solution of lye-water to put them in best possible condition. This precaution and the care to remove all entrained air will probably account for the higher rates of heat transmission at a velocity of 170 feet per minute than those recorded in Table II.

The steam pressure drops previously noted result from insufficient steam space and could be remedied by wider tube spacing and consequent increase in weight of header and shell. Unless such an increase of steam space were made, abnormally high back pressures for auxiliary machinery supplying the exhaust steam for feed heating would be necessary.

In a bulletin entitled "A Study in Heat Transmission,"* Clement and Garland discuss the condition on the steam side of a heater tube. The tests reported dealt with the heat transfer through the film to the metal wall, and the steam wall temperatures were given by thermo couples soldered into small holes in the surface of the tube. While such a study reveals many points of value it does not arrive at constants useful in design because the metal wall temperature is always an unknown factor. However, the conclusions of Messrs. Clement and Garland in regard to steam side are so much to the point that they are quoted verbatim: "On the steam side of the tube, if a condition of equilibrium is assumed to maintain, there will be the condition of saturated steam imparting its heat to a film of water on the surface of the tube, which film may be assumed to be of a constant thickness, and which is constantly being replaced or renewed by the condensation of fresh steam. The agitation of the steam in contact with the water film will, therefore, if it does not agitate this film, have no effect upon the rate of heat transmission or the conductance, at least, in the case of the present experiments; for the maximum velocity of approach of the steam towards the tube, due to the condensation, is about 1/5 foot per second. This would lead one to believe that the temperature on the surface of separation of the steam from the water film must be practically the temperature of the saturated steam. If such is the case, the agitation of the steam alone would not affect the conductance, as it would not change the temperature drop through the tube. The solution of the problem of more efficient heat transmission from steam to surface of tube, therefore, lies solely in the removal or agitation of the water film."

"It might be well to point out here that it is the rate of agitation and not necessarily the velocity of the water that produces the change in heat transmission. The velocity in feet per second has been used for the reason that it is the only definite quantity available to express this rate of agitation. It may be, to a certain extent, a measure of the rate of agitation,

*University of Illinois, Engineering Experiment Station, Bulletin No. 40.

but it is not necessarily so. The rate of agitation may be defined as the number of times per second that each particle of water comes into contact with the tube or with the film on the tube if we consider that the film is indestructible, although it is more likely that the film is in a process of continuous renewal. In the case of the present experiments, it is probable that if elaborate baffles had been placed in the tube, the heat transmission for the same velocity of flow would have been increased. Also, if the stream of water on entering the tube had been given a rotary motion the heat transmission would have been increased while the velocity through the tube remained constant."

In other words, in a feed heater the controlling resistance lies on the water side and is subject to correction by mechanical agitating or mixing devices up to the point where water ceases to flow as a uniform fluid, the point known as the critical velocity. The foregoing tests show how effective spiral retarders have proved in the Bureau type of heater.

At a water rate of 1,500 pounds per tube per hour the heat transmission was increased 16 per cent., at 2,000 pounds, 7.5 per cent., at 2,500 pounds, 14.9 per cent., and at 3,000 pounds, 14.8 per cent.

V.—Heat Transfer in Film Heaters. Schutte and Koerting Spirally-Corrugated Film Heater.

A form of heater in which the agitation is accomplished by a deformation of the tube in process of manufacture is the spirally-corrugated film heater manufactured by Schutte and Koerting. The heating element in this apparatus consists of two spirally-corrugated tubes, one within the other, the water path being the small clearance between the two. Thus the water is directed in a spiral path due to the corrugations, and for a given velocity the particles of water come more often in contact with the heating surface because they are contained within an annular space whose perimeter is large in comparison with its area.

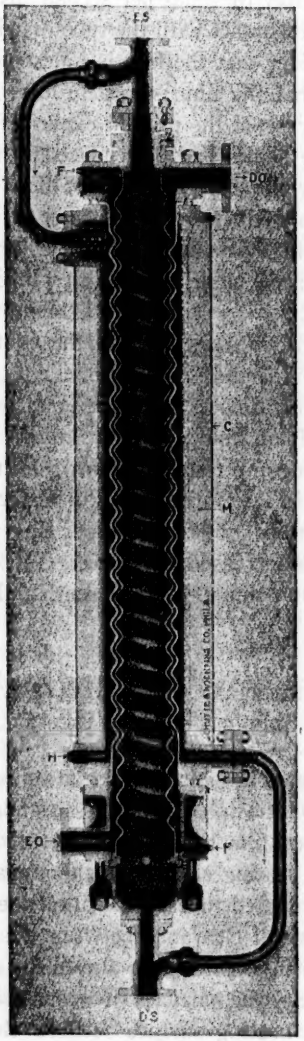


FIG. 8.

The first of these units to be tested as a feed-water heater is the same as the one whose performance as an oil heater is reported in the May, 1912, JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.* The heater, which is shown in figure 8, consists of a cylindrical composition casting containing one pair of spirally-corrugated copper tubes, one of which screws into the other. The water passes upward through the heater in a 7/64-inch nominal film between the tubes, the steam circulating downward around the outer tube and within the inner tube. The steam and water joints are external and the only failure which would not be immediately noticed would be that of a tube proper. The principal dimensions of the heater tested are given in Appendix I.

The series of tests proposed was with rates of water flow of 5,000, 10,000, 15,000 and 20,000 pounds per hour at steam pressures of 10 and 50 pounds per square inch gage. When the heater was put into service it was found that the least practicable rate of discharge was 7,500 pounds of water per hour, for at lower rates the outlet water was at such a high temperature that much of it was flashing into steam beyond the outlet control valve. Above 15,000 pounds per hour the friction drop was excessive, so the rates of discharge selected were 7,500, 10,000 and 15,000 pounds per hour.

The heat transfer was so rapid and the consequent supply of steam so great, that the steam pressures which could be used were limited by two conditions: firstly, the pressure had to be low enough that the corresponding temperature of inlet steam did not heat the water to the boiling point, and secondly, the pressure had to be high enough to keep the heater clear of water at the bottom, so as to obtain full benefit of all the heating surface and force the condensate to the weighing tanks. With atmospheric pressure at the condensed steam receiver, the inlet steam pressure varied from 11 to 31 pounds, indicating the adaptability of this apparatus with some modification to condenser service. The resistance through the

*Test of an External-Joint Film Oil Heater at the Engineering Experiment Station, Annapolis, Md., JOURNAL AM. SOC. NAV. ENG., 1912, p. 426.

heater was determined by a differential gage. For discharges below 12,000 pounds per hour the pressure drop is moderate, increasing from 7 pounds at 8,000 pounds of water per hour to nearly 40 pounds when the rate of flow is 15,000 pounds per hour.

The complete data from the tests is given in Table VI and the principal results shown graphically in figure 9. The heat transmitted in B.t.u. per square foot per hour reaches a value of 147,922 at the highest velocity. This corresponds to a heat transmission per square foot per hour per degree average temperature difference of 1,227 B.t.u. and per degree of mean temperature difference of 1,254 B.t.u., more than double that obtained at equivalent velocities in the Bureau type of heater. The true velocity of water is in this type problematical, as the spiral corrugation will produce an agitation greater than direct flow through a uniform film of the same area. In every case the axial velocity is considered, and the velocity corresponding to a reasonable friction loss would be about 250 feet per minute, or 12,930 pounds of water per hour.

Although the tests of this unit furnish no direct data to advance the theory developed in a previous section, the tests have been cited to show the high rates of heat transfer which may be attained with proper water agitation, as well as to form an introduction to more extensive tests of a similar type of heater.

The film type of heater had been selected by the Bureau of Steam Engineering for installation on the U. S. S. *Oklahoma,* conditional upon a maximum pressure drop of 10 pounds at full load. The builders thereupon constructed a small experimental heater after the design for the *Oklahoma,* containing four double-tube elements instead of sixty-six in each of two heaters for the ship, and sent the same to the Engineering Experiment Station, Annapolis, Md., where it was tested under the writer's direction in 1912. These tests covered a wide range of water velocities, steam pressures, tube arrangement and water temperatures, and the results will be presented in

TABLE VI.—DATA AND RESULTS OF TESTS ON SCHUTTE AND KOERTING FILM-OIL HEATER WHEN USED AS A FEED-WATER HEATER.

1. Test number	1	2	3	4
2. Date, May, 1912	25	25	25	27
3. Duration of test, hours	1	1	1	1
4. Barometer, ins. of mercury	29.96	29.96	29.96	30.04
5. Temperature of room, °F.	84.7	88.0	88.0	79.7
6. Water through heater, lbs. per hour	7,780.5	10,897.5	11,985.5	14,807.5
7. Pressure at inlet, lbs. gage	244.8	247.6	244.7	249.2
8. Pressure at outlet, lbs. gage.	238.3	234.7	230.8	215.9
9. Differential gage reading, inches of mercury	14.3	28.3	30.6	...
10. Resistance through heater, pounds per square inch	6.5	12.9	13.9	33.3
11. Temperature at inlet, °F.	73.0	72.4	72.4	72.1
12. Temperature at outlet, °F.	194.8	184.9	186.5	190.5
13. Temperature rise, degs. F.	121.8	112.5	114.1	118.4
14. Mean specific heat	.988	.988	.988	.988
15. Heat absorbed by water, B.t.u. per hour	936,293	1,211,257	1,351,135	1,732,170
16. Steam condensed, lbs. per hour	931.0	1,229.3	1,382.5	1,830.3
17. Steam pressure at inlet, lbs. gage	11.0	16.6	19.9	27.2
18. Temperature at inlet, °F.	307.5	278.7	299.1	281.2
19. Temperature corresponding to absolute pressure at inlet, degrees F	241.5	252.7	258.6	270.2
20. Superheat, degrees F.	66.0	26.0	40.5	11.0
21. Temperature of condensed steam at outlet, degrees F.	211.4	206.7	212.7	222.5
22. Equivalent dry and saturated steam, lbs. per hour.	956.7	1,243.2	1,406.7	1,839.8
23. Heat supplied by steam, B.t.u. per hour	943,196	1,232,619	1,390,380	1,803,212
24. Radiation and unaccounted losses, B.t.u. per hour	7,053	21,776	39,737	72,502
25. Velocity of water through tube, feet per minute	152.8	213.7	235.1	290.6
26. Transmission in B.t.u. per square foot per hour	79,957	103,438	115,383	147,922
27. Average arithmetical temperature difference, °F.	125.6	114.1	126.5	120.6
28. Transmission in B.t.u. per sq. foot of heating surface per deg. F. of average arithmetical temperature difference per hour	636.6	906.6	912.1	1,226,6
29. Mean temperature difference, degrees F.	125.1	112.8	126.0	117.9
30. Heat transmission in B.t.u. per square foot of heating surface per degree F. mean temperature difference per hour	639.1	917.0	915.7	1,254.6
31. Dry steam consumed per sq. foot of heating surface per hour, pounds	81.69	106.16	120.13	157.11
32. Water heated per sq. foot of heating surface per hour, pounds	664.9	930.6	1,023.5	1,264.5
33. Water heated per pound of dry steam, pounds	8.133	8.766	8.520	8.048

KEUFFEL & ESSER CO NEW YORK, NO. 334 A.
WATER PER Sq FT OF H.S. PER HR.-POUNDS, BTU PER Sq FT OF H.S. PER HR. PER DEGREE FAHR. TEMPERATURE DIFF
400 500 600 700 800 900 1000 1100 1200 1300
8 9 10 11 12 13 14 15
WATER HEATED PER HOUR-THOUSANDS OF POUNDS.
THOUSANDS OF B.T.U PER Sq FT OF H.S PER HR.
60 80 100 120 140 160
PRESSURE DROP THRU HEATER-LB PER Sq IN.
0 01 02 03 04 05 06 07 08 09

FIG. 9.

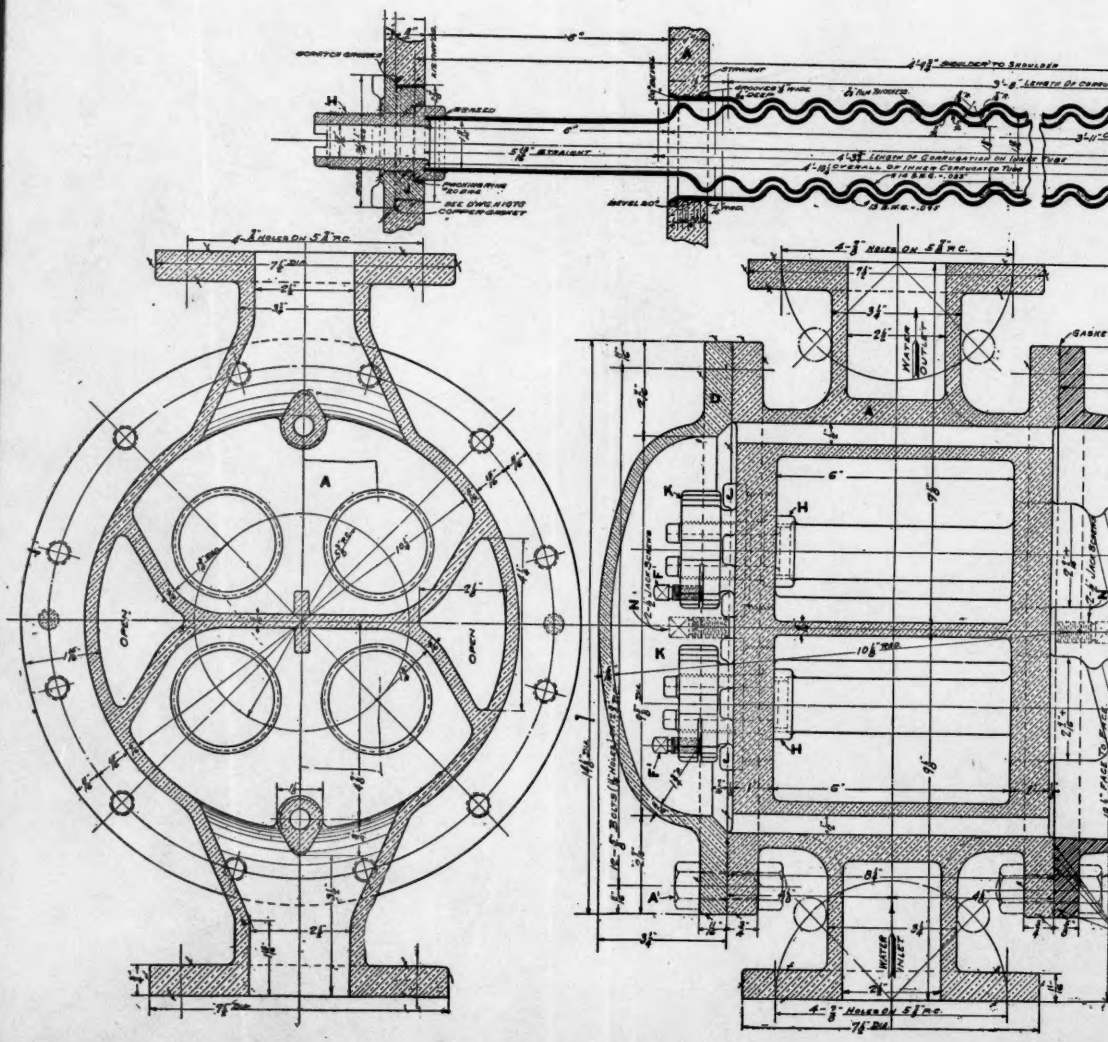
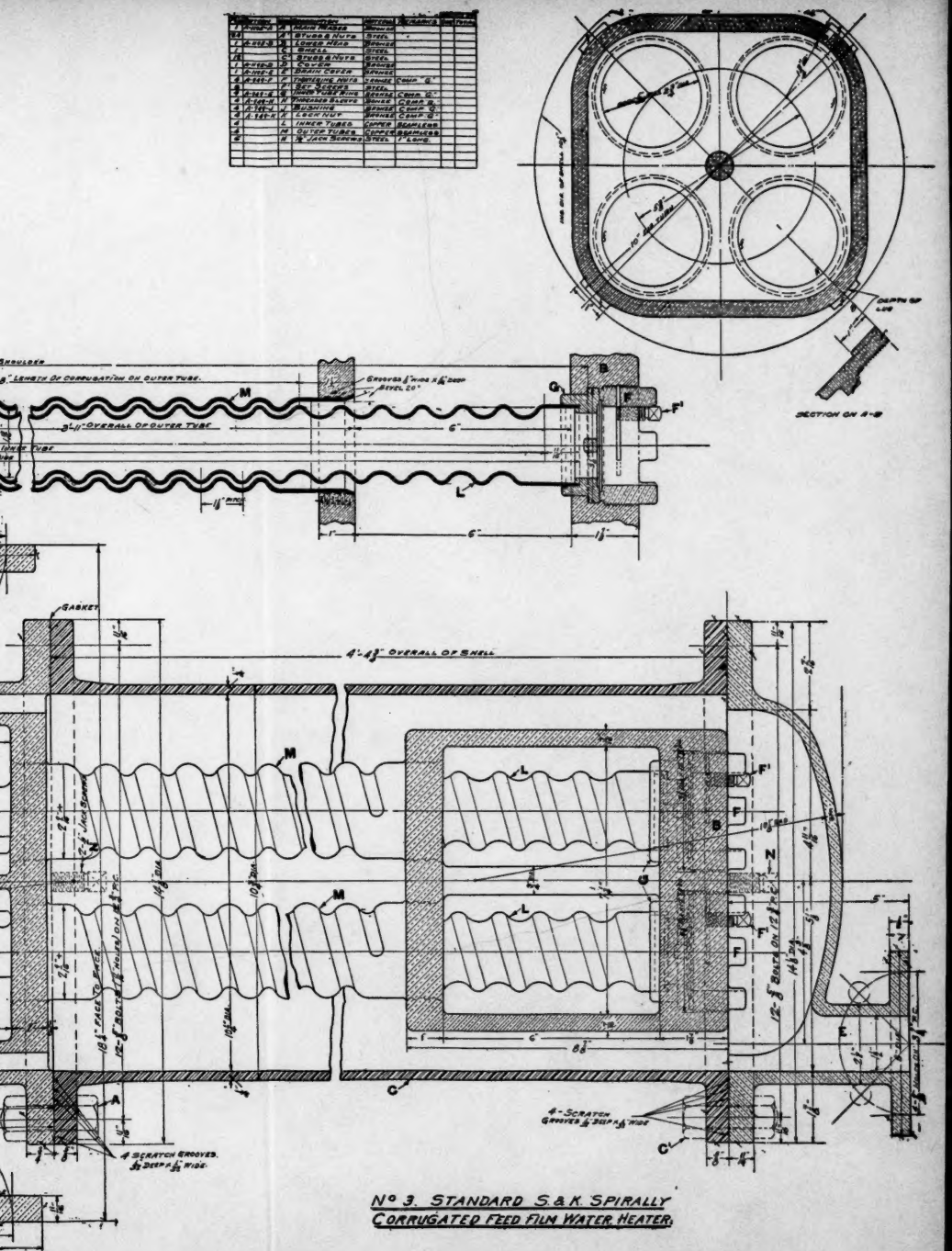


PLATE II.



No 3. STANDARD S & K SPIRALLY CORRUGATED FEED FILM WATER HEATER.

ATE II.

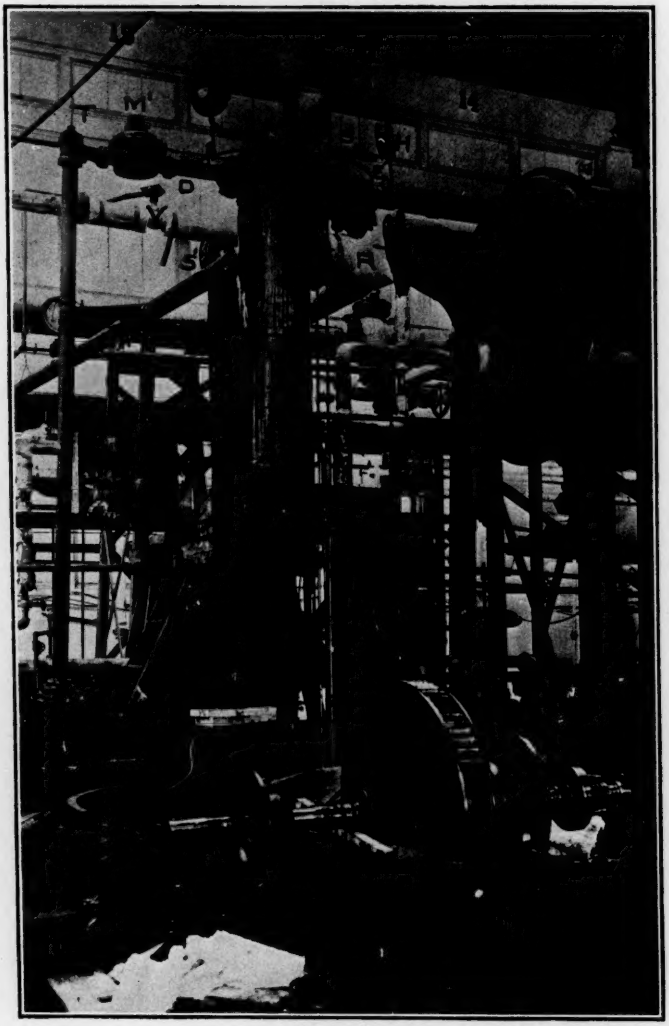


FIG. 10.

SCHUTTE AND KOERTING EXPERIMENTAL FEED-WATER HEATER ERECTED FOR TEST.

detail as containing a mass of information on this particular type of apparatus.

The experimental apparatus is shown in Plate II. There are four pairs of spirally-corrugated tubes arranged in two passes. As originally constructed the approximate water-film thickness was 7/64 inch, but there were supplied four additional inner tubes with which to replace those in the heater and obtain a film thickness of approximately 3/16 inch. The heater was apparently designed for horizontal installation, but a slight modification of the lower header permitted of vertical mounting.

The heater details give the outer tubes as No. 13 B.W.G., 3 feet 11 inches long, rigidly expanded into the headers. The inner tubes are No. 14 B.W.G., 4 feet 10¼ inches long overall and are screwed into place, the joints being made tight by locking nuts and copper packing rings. Water enters the main header through a 2½-inch opening, is directed by a center baffle down two pairs of tubes to the lower head and then upward and out. Steam connection is made through a 7-inch flange, and the condensation is removed by a 1½-inch outlet in the drain cover. Suitable lagging is provided. The principal dimensions and calculated areas of steam heating surface are given in Appendix II.

The apparatus was erected for vertical test as shown in figure 10. The water to be heated is delivered under pressure past the inlet water thermometer T, the water meter M′, and the inlet pressure gage G. Discharge pressure, maintained at about 200 pounds, is taken at gage H and discharge temperature by thermometer S. To measure the friction drop a differential mercury gage was connected across the feed line at D and E.

Steam pressure was controlled by valve V and its initial condition shown by thermometers and pressure gage attached near R. Pressures were also taken within the shell, and the temperature of condensed steam by a thermometer immersed in a drain pot at outlet. Accumulated air was withdrawn by

a pet cock in the drain pot and the water of condensation was cooled and weighed, thus obtaining a measure of the heat input as well as heat imparted to the water.

The object of the tests was to determine the effect on heat transfer of a number of variables:

(1) Velocity of feed water through the water film;
(2) Steam pressure;
(3) Temperature of inlet feed;
(4) Variation in mean film thickness;
(5) Vertical and horizontal installation.

The data and results of tests are contained in Tables VII to XI. In each of these tables the individual trials are tabulated in groups of ascending water rates at steam pressures of approximately 5, 10 and 15 pounds gage pressure. The several tables cover the following conditions:

VII. Vertical installation, 7/64-inch film thickness, inlet water temperature 66.5 degrees to 72.2 degrees F., water rates 1,750 to 10,950 pounds per tube per hour;
VIII. Horizontal installation, 7/64-inch film thickness, inlet water temperature 52.5 degrees to 55.8 degrees F., water rates 2,330 to 10,200 pounds per tube per hour;
IX. Vertical installation, 3/16-inch film thickness, inlet water temperature 70.2 degrees to 71.1 degrees F., water rates 2,640 to 10,310 pounds per tube per hour;
X. Vertical installation, 3/16-inch film thickness, inlet water temperature 87.6 degrees to 94.4 degrees F., water rates 2,660 to 12,860 pounds per tube per hour;
XI. Horizontal installation, 3/16-inch film thickness, inlet water temperature 88.6 degrees to 95.3 degrees F., water rates 2,520 to 12,220 pounds per tube per hour.

The sources of the data for the various columns of the foregoing tables and the method of obtaining the deduced data are shown in Appendix III.

TABLE VII.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER.—TABLE OF DATA AND RESULTS.
7/64-INCH WATER FILM.—VERTICAL ARRANGEMENT.

Water Data.															Steam Data.												Reduced Data.									
Number of Test.	Date, November, 1912.	Duration of Test. Minutes.	Barometer. Inches of Mercury.	Room Temperature. Degrees F.	Water Pumped through Heater. Pounds per Hour.	Temperature at Inlet. Degrees F.	Temperature at Outlet. Degrees F.	Temperature Rise. Degrees F.	Density of Water.	Specific Heat of Given Water.	B.T.U. Absorbed by Water per Hour.	Friction Drop. Inches of Mercury.	Water Pressure at Inlet. Pounds per Sq.In.	Water Pressure at Outlet. Pounds per Sq.In.	Friction Drop. Pounds per Sq.In.	Pressure at Inlet. Pounds Gage.	Pressure at Inlet. Pounds Absolute.	Temperature corresponding to Pressure.	Degrees Superheat.	Temperature at Outlet. Degrees F.	Temperature of Condensed Steam. Degrees F.	Heat Subducted per Pound. B.T.U.	Steam Condensed per Hour. Pounds.	B.T.U. Subducted by Steam per Hour.	Radiation. B.T.U. per Hour.	Equivalent Dry and Saturated Steam. Pounds per Hour.	Adiat Velocity. Feet per Minute.	B.T.U. Transmitted per Sq.Ft. per Hour.	Average Log Mean Temperature Difference.	Mean Temperature Difference.	B.T.U. per Sq.Ft. per Hour per Degree Mean Temp. Difference.	B.T.U. per Sq.Ft. per Hour per Degree Mean Temp. Difference.	Pounds of Steam per Hour per Square Foot of Water Surface.	B.T.U. Transmitted per Hour per Foot Length.	Efficiency, as a Ratio.	Pounds of Water per Tube per Hour.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38
37	18	30	30.14	64.1	4727	55.4	198.2	142.8	1.008	.970	653380	5.35	225.4	213.1	2.34	4.73	19.53	308.5	226.7	74.8	224.1	221.6	1001.2	682	682818	25436	706.6	120.8	30630	99.9	79.6	306.6	384.8	149.8	221.1	95.7	2587
38	18	30	30.12	70.0	9672	54.7	180.1	125.4	1.008	.970	1176476	18.67	201.9	193.4	8.49	4.70	19.49	308.5	226.6	76.9	225.6	225.8	998.9	1147	1145738	-30740	1391.7	247.8	55193	109.2	95.9	505.1	575.1	183.2	453.4	----	4636
39	18	30	30.14	72.0	14993	54.7	167.6	112.9	1.008	.970	1643927	----	227.5	209.4	18.1	5.45	20.25	265.5	228.7	36.8	229.3	226.8	976.7	1702	1662344	20417	1732.6	364.1	76974	117.5	107.9	655.1	713.4	115.6	702.9	98.8	7496
40	19	30	30.06	70.0	20402	54.6	153.0	98.4	1.008	.970	1947386	----	237.9	206.9	31.0	5.45	20.21	269.9	228.5	41.4	228.1	227.8	979.9	2025	1984398	36972	2065.5	522.6	91851	124.7	117.9	732.1	774.3	101.2	956.4	98.1	10201
41	18	30	30.14	65.9	4729	55.2	213.8	158.6	1.008	.970	797514	5.09	210.3	208.0	2.32	10.12	24.92	310.4	239.9	70.5	239.2	237.7	988.0	751	741988	14474	777.3	186.1	34106	105.4	81.0	323.6	421.1	164.4	221.7	98.0	2365
42	18	30	30.15	70.3	9572	54.7	193.1	138.4	1.008	.970	1885016	17.57	215.9	207.7	8.16	10.50	25.28	289.3	240.7	48.6	242.0	240.7	974.8	1432	1395114	110898	1466.4	345.2	60842	116.8	101.5	515.8	593.5	153.2	448.7	----	4706
43	19	30	30.09	69.1	14427	55.8	174.3	118.5	1.008	.970	1658313	37.22	227.1	206.2	16.94	10.47	25.25	286.0	240.6	55.4	240.3	238.4	980.3	1750	1715586	97213	1799.0	369.6	77460	125.5	115.6	617.2	670.1	124.7	676.3	96.7	7213
44	19	30	30.06	69.0	19600	53.5	159.6	106.1	1.008	.970	2027173	----	237.0	209.2	31.8	9.83	24.59	277.5	239.2	38.3	239.4	238.9	971.1	2136	2074870	57097	2178.7	502.0	94565	132.6	125.2	713.2	755.3	111.2	918.8	97.2	9800
45	18	30	30.14	68.0	4732	55.5	222.2	166.7	1.008	.970	774855	5.80	205.2	202.6	2.64	14.75	29.55	311.4	249.5	61.9	249.1	247.4	977.0	786	767922	-6333	810.4	182.8	36395	110.6	85.0	386.4	427.4	169.1	224.6	----	2396
46	18	30	30.14	71.3	10339	54.8	198.8	144.0	1.008	.970	1444147	20.36	210.6	201.2	9.35	15.16	29.96	275.2	250.2	25.0	250.1	249.3	957.7	1542	1476774	38627	1560.9	364.8	67701	123.4	107.8	548.6	628.0	150.3	484.7	97.8	5170
47	19	30	30.09	68.0	14452	54.6	183.4	128.8	1.008	.970	1809570	36.42	221.0	204.4	16.57	15.41	30.19	271.6	250.7	20.9	250.3	250.2	955.4	1931	1847744	42174	1995.3	370.2	84645	131.7	120.4	648.7	703.0	135.3	677.5	97.7	7226
48	19	30	30.01	71.7	19469	52.5	165.4	112.9	1.008	.970	2135231	----	234.9	204.9	30.0	15.20	29.74	279.7	249.8	29.9	250.0	249.7	959.9	2284	2198412	57021	2318.3	499.4	100107	140.8	132.9	711.0	759.3	118.3	914.1	97.4	9750

TABLE VIII.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER. TABLE OF DATA AND RESULTS.
7/64-INCH WATER FILM.—HORIZONTAL ARRANGEMENT.

TABLE IX.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER. 3/16-INCH WATER FILM.—VERTICAL ARRANGEMENT.

Number of Test.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38
Water Data.																																						
Date — 1911.	2	10	10	10	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11
Duration of Test.	3	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56	56
Barometer — Inches of Mercury.	4	29.89	30.06	30.01	30.06	30.06	30.07	30.07	30.07	30.07	30.07	30.07	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87	29.87
Room Temperature, Degrees F.	5	73.0	81.3	81.0	81.0	81.0	76.7	71.1	68.5	71.1	76.5	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1
Steam Data.																																						
River Water Pumped through Heater. Pounds per Hour.	6	53281	16659	15373	28627	5979	10760	11554	11554	11554	5550	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554	11554
Temperature at Entrance to Tubes, Degrees F.	7	70.1	70.2	70.2	71.1	71.1	70.3	70.2	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1	71.1
Temperature at Outlet from Tubes, Degrees F.	8	207.9	203.0	176.7	166.8	220.1	202.6	188.5	176.1	176.1	195.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1	176.1
Temperature Rise, Degrees F.	9	137.8	122.8	105.5	85.7	149.4	131.9	117.3	105.1	105.1	124.0	105.1	105.1	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7	104.7
Speed of Screw.	10	1.0074	1.0074	1.0074	1.0074	1.0074	1.0074	1.0074	1.0077	1.0077	1.0074	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077	1.0077
Specific Heat of Water.	11	.9720	.9720	.9720	.9720	.9720	.9720	.9720	.9714	.9714	.9720	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714	.9714
B.T.U. Absorbed by Water per Hour.	12	73282	19722	16238	23772	7454	19360	13560	20703	20703	6720	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703	20703
Reduction Zero. Inches of Mercury.	13	2.76	3.12	2.70	2.44	2.70	2.56	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76	2.76
Reduction Pressure at Inlet, Pounds per Sq. In.	14	1053.7	1039.6	1037.5	1037.7	1035.7	1035.0	1035.4	1035.6	1035.4	1035.6	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4	1035.4
Reduction Drop, Pounds per Sq. In.	15	132.4	119.4	117.4	114.2	115.7	115.4	115.0	116.1	115.4	115.6	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4	115.4
Reduction Drop, Pounds per Sq. In.	16	1.27	4.15	8.06	13.50	4.25	7.57	11.26	8.74	1.16	14.56	6.11	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62
Pressure at Inlet, Pounds per Sq. In.	17	1.56	4.17	8.06	13.50	4.68	9.00	11.38	8.70	1.23	14.56	6.23	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62	14.62
Pressure at Intake, Pounds Above Zero.	18	1029	1046	1045	1037	1029	1028	1024	1027	1029	1021	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029	1029
Temperature at Inlet, Degrees F.	19	361.6	361.0	360.8	360.1	360.5	359.3	358.8	359.6	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4	359.4
Temperature of Condensed Steam, Degrees F.	20	286.1	286.5	286.5	286.7	286.1	286.2	286.2	286.1	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2
Reduction Superheat.	21	16.5	34.5	34.8	16.5	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1
Temperature at Inlet, Degrees F.	22	302.6	251.4	251.4	251.4	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1	295.1
Temperature of Condensed Steam, Degrees F.	23	286.2	286.5	286.5	286.7	286.1	286.2	286.2	286.1	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2	286.2
Heat Absorbed per Pound Steam, B.T.U.	24	350.2	1001.3	990.6	993.2	994.5	991.8	989.6	990.3	990.3	989.6	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3	990.3
Steam Condensed per Hour.	25	745.0	1995.5	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0	1960.0
B.T.U. Transmitted per Hour.	26	74859	1997.84	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00	1960.00
Heating Surface.	27	21440	21915	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187	21187
Total Transmission per sq. ft. Heating Surface per Hour.	28	768.8	1368.4	1729.7	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3	1852.3
Axial Velocity, Feet per Minute.	29	50.0	127.9	256.6	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3	370.3
B.T.U. Transmitted per Sq. Ft. per Hour.	30	25661	68211	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855	87855
Average Arithmetical Temperature Difference.	31	87.1	34.9	29.6	23.7	47.4	43.2	40.8	33.9	33.9	39.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9	33.9
Mean Temperature Difference.	32	64.1	71.7	91.2	100.2	69.7	72.0	74.8	82.1	82.1	73.2	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1	82.1
Heating Surface per Pound Steam per Hour.	33	405.4	685.6	771.7	870.6	396.0	648.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6
B.T.U. Transmitted per Square Foot Heating Surface per Hour.	34	333.0	760.6	865.0	935.8	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2	1056.2
Square Feet of Heating Surface per 1000 Pounds of Water Heated.	35	1482.5	1316.6	1158.5	954.8	948.1	1542.6	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0	1417.0
Water Entered per Square Foot Heating Surface per Hour.	36	450.2	731.7	863.9	1048.6	1052.6	648.2	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6	705.6
Efficiency on Basis of Test.	37	97.1	96.1	96.9	98.1	96.4	97.5	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6	97.6
Pounds of Water per Foot per Hour.	38	4996	5559	7487	10553	8440	7076	10050	8075	7950	10553	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450	7450

Water Data.																Steam Data.										Technical Data.									
Number of Test	Date — 1911.	Duration of Test. Hours.	Barometer. Inches of Mercury.	Steam Gauge Pressure. Pounds per Square Inch.	Water Meter. Pounds per Hour.	Temperature of Inlet. Degrees F.	Temperature of Outlet. Degrees F.	Temperature Rise. Degrees F.	Water Meter. Factor.	Weight of Water. Pounds per Cubic Foot.	B.T.U. Delivered by Water per Hour.	Friction Drop. Inches of Mercury.	Water Pressure. Gauge. Pounds per Square Inch.	Friction Drop. Pounds per Square Inch.	Pressure of Inlet. Pounds Gauge.	Pressure of Inlet. Pounds Absolute.	Temperature of Inlet. Degrees F.	Temperature Corresponding to Pressure. Degrees F.	Degrees Superheat.	Temperature of Outlet of Steam. Degrees F.	Temperature of Condensation. Degrees F.	Heat Delivered per Hour. B.T.U.	Steam Condensed per Hour. Pounds.	B.T.U. Delivered per Hour.	Radiation. B.T.U. per Hour.	Difference, BT.U. per Hour. Radiation and Loss.	Axial Velocity. Feet per Minute.	B.T.U. Transmitted per Square Foot per Hour.	Average Arithmetical Temperature Difference.	Mean Temperature Difference.	B.T.U. per Sq.ft. per Hour per Degree Average Temperature Difference.	B.T.U. per Sq.ft. per Hour per Degree Mean Temperature Difference.	Ratio of Tube Length to Diameter.	Water passed per Sq.ft. of H.S. per Hour.	Efficiency of Heater.	Pounds of Water per Pound of Coal.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38
65	26	30	30.27	61	5660	68.9	207.5	138.6	1.0079	.974	677061	3.21	204.8	203.3	1.46	4.91	19.77	309.1	227.3	81.8	227.2	225.7	1000.8	707	707566	30656	735	98.1	73108	79.1	61.0	418.6	542.8	125.4	256.7	95.7	2931
66	26	30	30.19	65	9964	90.5	198.8	108.3	1.0079	.974	1051202	7.30	211.2	207.9	3.38	4.93	19.76	311.5	227.3	84.2	227.9	227.6	1000.0	1074	1074000	22798	1118	166.7	51404	62.6	49.2	622.3	742.8	112.2	487.2	97.9	4962
67	26	30	30.17	69	14525	83.5	187.1	97.6	1.0079	.974	1499658	14.59	213.5	206.9	6.64	4.76	19.87	239.8	226.9	12.9	228.2	227.5	966.2	1496	1445435	35977	1595	246.1	62942	88.6	76.8	778.1	874.9	101.5	724.9	97.5	7413
68	27	30	29.87	64	19137	90.3	178.7	88.4	1.0079	.974	1647636	22.60	214.3	204.0	10.29	4.80	19.87	267.7	225.9	41.8	226.5	226.6	980.5	1721	1687441	39745	1757	320.3	80973	91.4	83.8	881.5	961.5	91.8	935.8	97.6	9569
69	27	30	29.45	65	24850	94.0	171.3	77.3	1.0073	.974	1863675	36.28	218.0	201.6	16.40	4.83	19.89	279.6	226.0	53.6	226.1	225.8	986.9	1945	1919521	55046	1996	414.5	91134	93.3	87.8	976.8	1038.0	80.6	1210.3	97.1	12375
70	26	30	30.23	66	9680	90.4	218.7	128.3	1.0073	.974	712000	2.90	455.3	204.0	1.32	5.89	24.74	311.2	239.5	71.7	239.0	238.9	988.2	759	750044	42044	786	99.1	38719	84.9	69.1	406.9	533.3	138.4	277.8	94.7	2640
71	26	30	30.19	68	10304	88.1	206.2	118.1	1.0073	.974	1154960	7.71	212.3	208.8	3.51	5.83	24.66	296.6	239.3	57.3	239.8	238.9	960.2	1833	1814468	49598	1875	172.4	57945	92.1	77.7	629.2	745.8	123.7	503.9	97.6	5152
72	27	30	29.51	63	15958	90.8	194.5	103.7	1.0073	.974	1541058	15.19	210.8	203.9	6.92	5.76	24.25	307.9	238.4	69.5	238.9	238.4	987.2	1939	1916533	37475	1955	255.4	75358	95.7	85.5	787.4	881.4	100.5	746.1	97.6	7629
73	27	30	29.43	65	19387	87.6	184.8	97.2	1.0073	.974	1837949	23.10	214.7	204.2	10.52	5.73	24.18	293.6	238.2	55.4	238.6	238.5	966.0	1947	1880808	44859	1972	324.3	89778	102.0	93.7	880.2	958.1	101.7	948.0	97.6	9694
74	27	30	29.47	65	24887	90.7	178.3	87.6	1.0073	.974	2118396	36.54	224.1	207.5	16.64	5.72	24.19	268.1	238.2	29.9	238.0	237.7	968.1	2247	2175521	56725	2281	415.7	105600	103.7	97.2	999.0	1065.8	91.8	1814.5	97.6	12419
75	26	30	30.21	66	5313	94.0	230.8	136.8	1.0073	.974	707692	2.58	203.9	202.7	1.17	14.93	29.76	383.6	249.8	63.8	250.4	249.0	976.1	760	742444	34758	784	89.0	34666	87.4	65.0	395.9	538.4	147.6	259.8	95.3	2657
76	26	30	30.18	68	10260	88.1	214.0	125.9	1.0073	.974	1257876	7.65	214.1	210.6	3.48	15.11	29.93	395.3	250.2	44.1	250.6	249.1	948.1	1369	1297949	40073	1372	171.7	61510	99.1	84.0	620.7	732.1	133.7	501.7	96.9	5130
77	27	30	29.47	64	15450	92.2	201.9	109.7	1.0073	.974	1650060	15.19	210.5	203.6	6.92	15.01	29.68	372.4	249.3	23.1	249.5	248.5	957.7	1765	1690341	40281	1786	258.6	80668	102.2	91.6	789.5	880.9	115.6	775.5	97.6	7725
78	27	30	29.44	65	19694	94.4	195.9	101.5	1.0073	.974	1948737	23.78	214.1	203.3	10.80	15.03	29.49	395.4	249.3	56.1	249.1	248.9	974.2	2057	2003989	56192	2137	329.8	95244	104.1	95.3	914.9	993.4	107.5	963.0	97.2	9847
79	27	30	29.49	65	25716	91.8	183.9	92.1	1.0073	.974	2396725	38.53	223.8	206.2	17.55	15.00	29.48	363.8	249.3	34.5	249.9	248.9	962.8	2463	2371376	64651	2507	430.5	112799	111.4	104.8	1012.6	1076.3	97.5	1257.5	97.3	12858

TABLE X.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER. TABLE OF DATA AND RESULTS.—3/16-INCH WATER FILM.—VERTICAL ARRANGEMENT.

TABLE XI.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER. TABLE OF DATA AND RESULTS.

3/16-INCH WATER FILM.—HORIZONTAL ARRANGEMENT.

Attempts were made to obtain the temperature of the water at the reversal of flow in the lower head. No practical way was found to get an accurate measure of this temperature, hence a temperature gradient could not be constructed, and the value of n, the exponent in the heat transfer relation, had to be determined indirectly. Inasmuch as the temperature rise on test was usually in excess of 100 degrees, the average temperature difference would not be close enough to the mean temperature to obtain the value of the exponent directly from a logarithmic plot. The mean temperature difference for all tests was computed for assumed values of n from 0.7 to 1.0 and the resulting values of K compared. It was found that for a value of $n = 0.9$, the values of K for one water velocity and varying steam and inlet-water temperatures were substantially constant. Hence for $n = 0.9$ the relation

$$Q = KS(t_s - t)^n$$

was satisfied, and for the film-heater heat transfer is proportional to the 0.9 power of the temperature difference. The values of K for 3/16-inch film and various water rates were determined as:

Water per tube, pounds per hour.	Coefficient, K.
2,500	706
5,000	1,049
7,500	1,460
10,000	1,493

The above illustrates clearly that slight advantage can be gained from water-film agitation above 7,500 pounds per tube per hour. This must be considered as the critical velocity for such construction, and higher rates would produce no increased heat transfer but would result in excessive friction losses.

The general test results were originally based on logarithmic mean temperature difference, and these are cited here as

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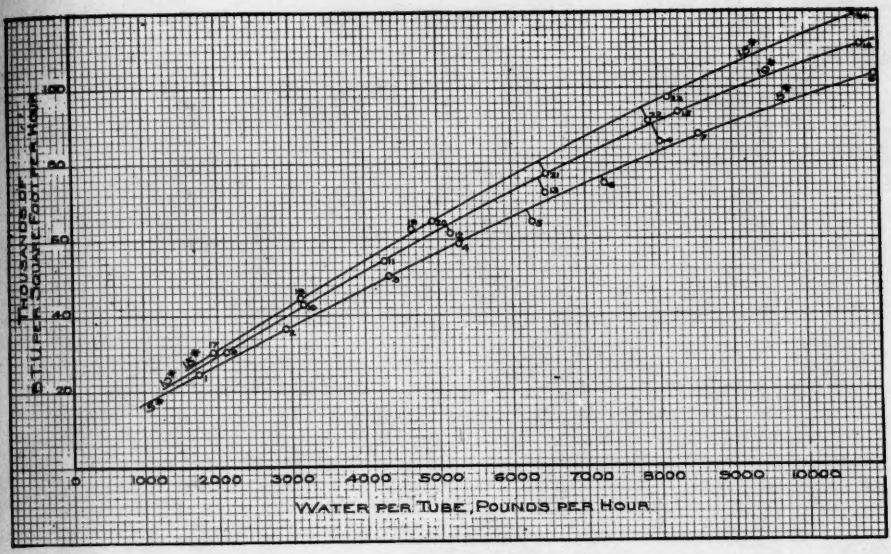
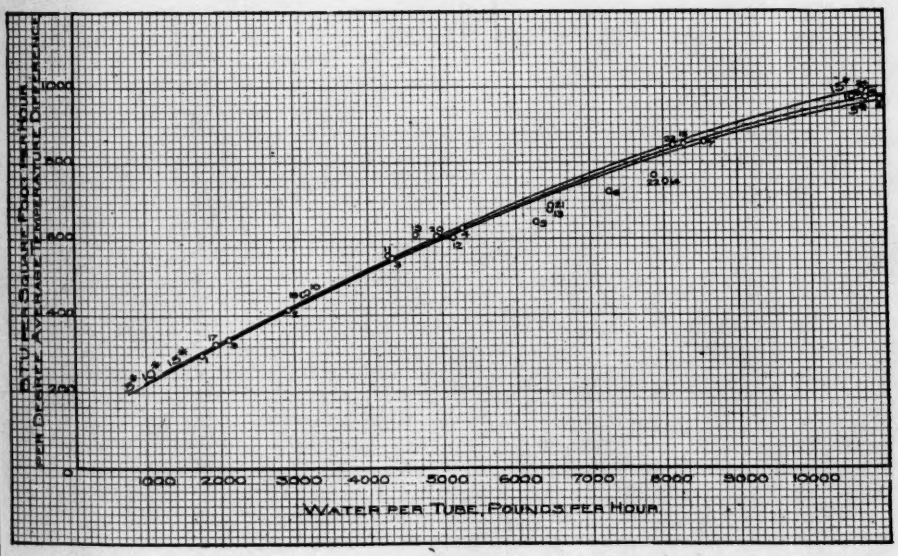
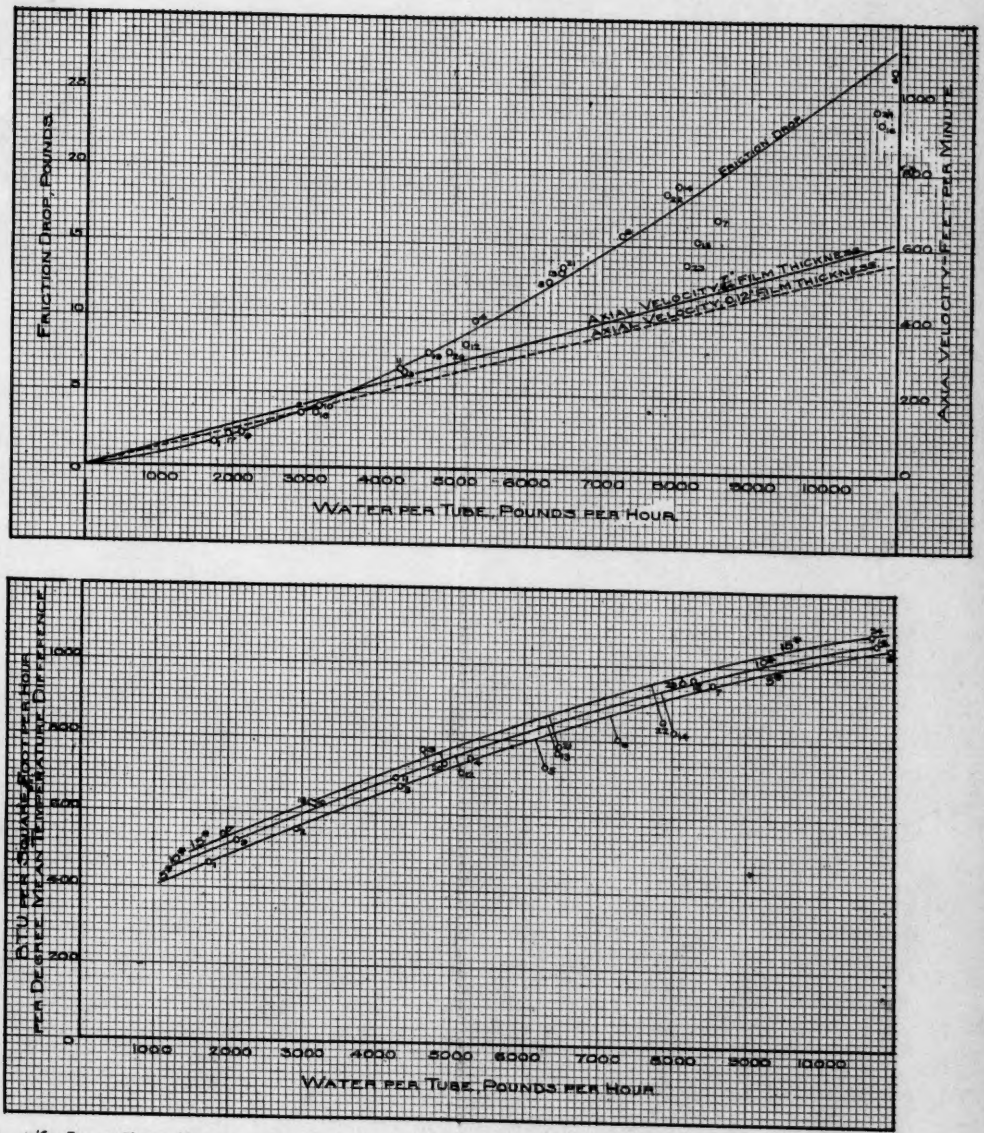



B.T.U. per Square Foot per Hour for Degree Average Temperature Difference
Water per Tube, Pounds per Hour
B.T.U. per Square Foot per Hour per Degree Mean Temperature Difference

CURVES FOR TEST 285.—SCHUTTE-KOERTING FEED-WATER HEATER. VERTICAL ARRANGEMENT. 7/64-

NOTE.—Small numerals indic

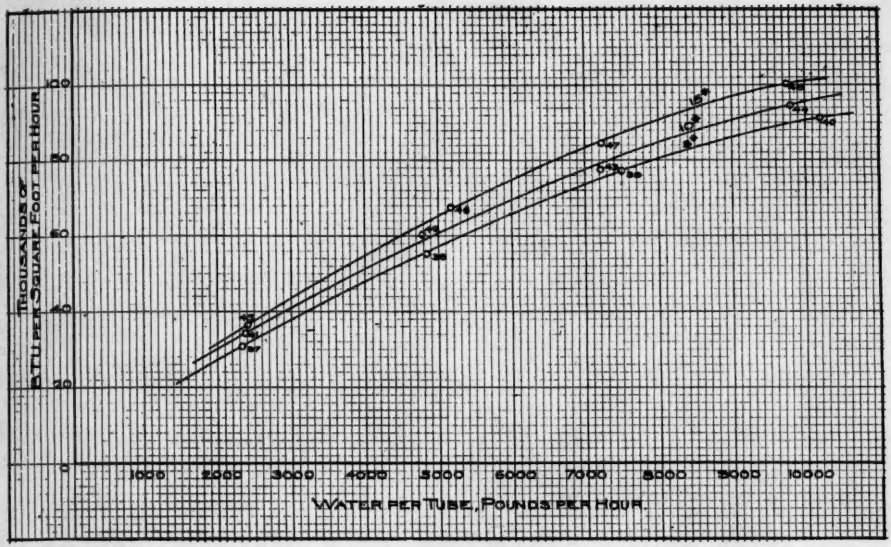
FIG. 11.

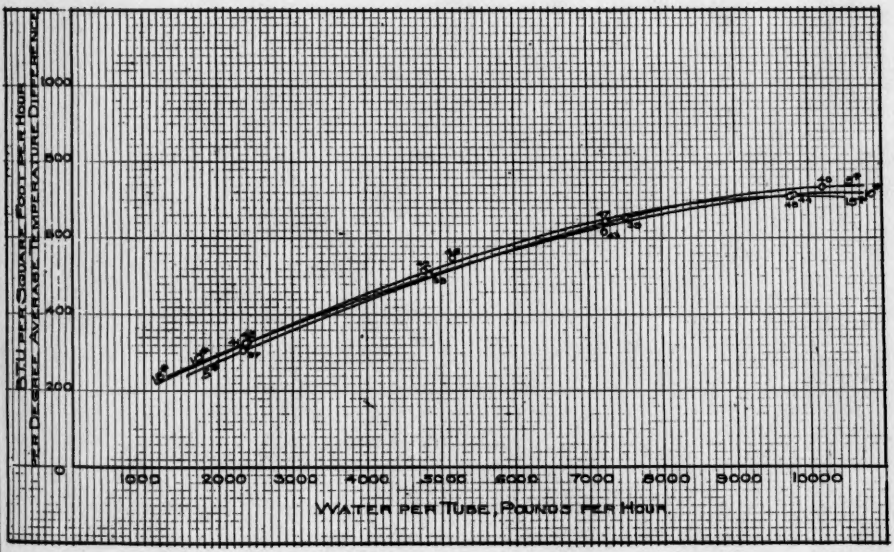


NT. 7/64-INCH FILM THICKNESS. U. S. NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MD.

nerals indicate number of run.

FIG. 11.

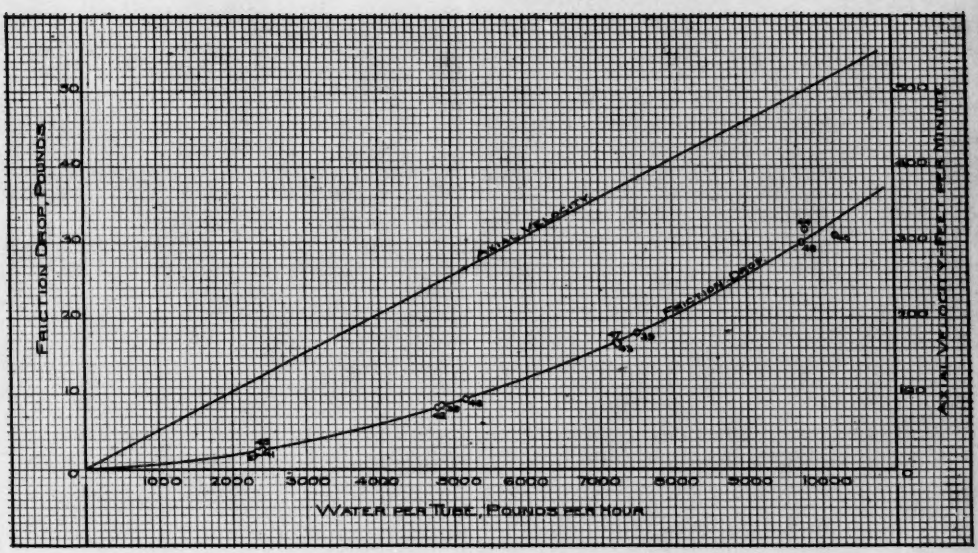


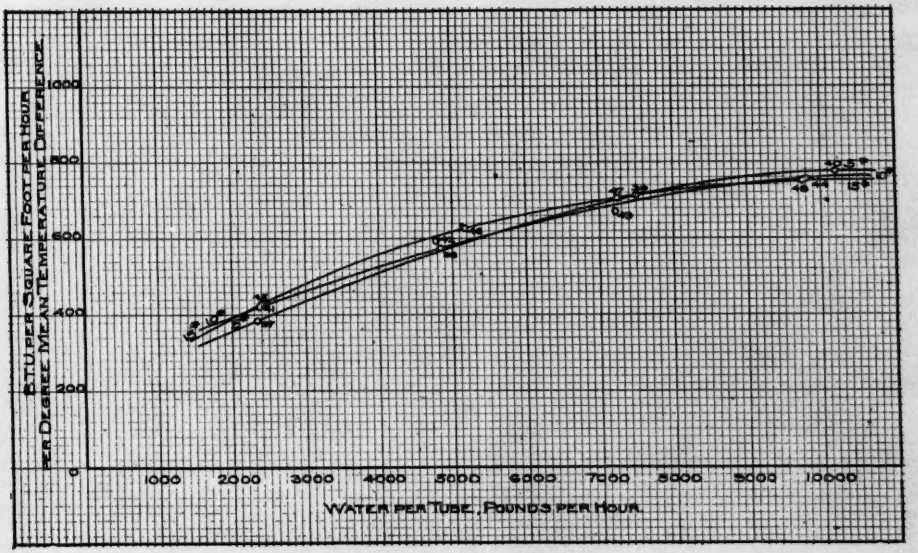


CURVES FOR TEST 285.—SCHUTTE-KOERTING FEED-WATER HEATER. HORIZONTAL ARRANGEMENT.

NOTE.—Small numerals in

FIG.

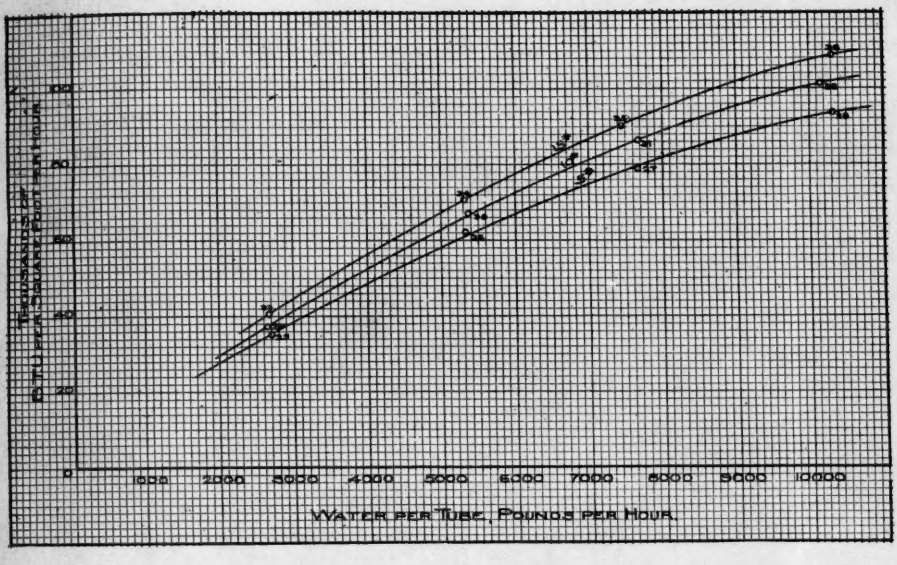


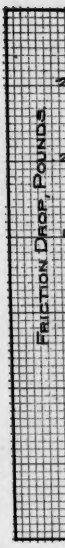


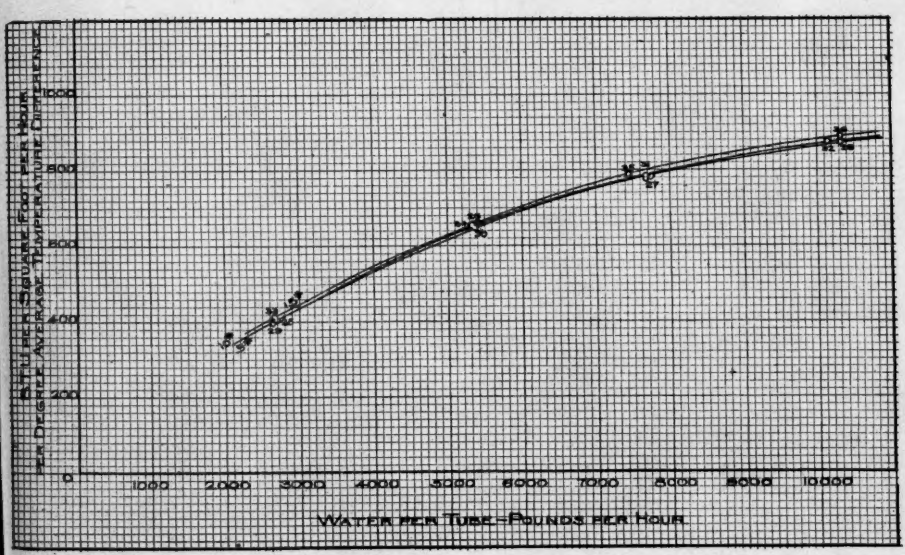
GEMENT. 7/64-INCH FILM THICKNESS. U. S. NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MD.

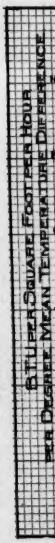
ll numerals indicate number of run.

FIG. 12.





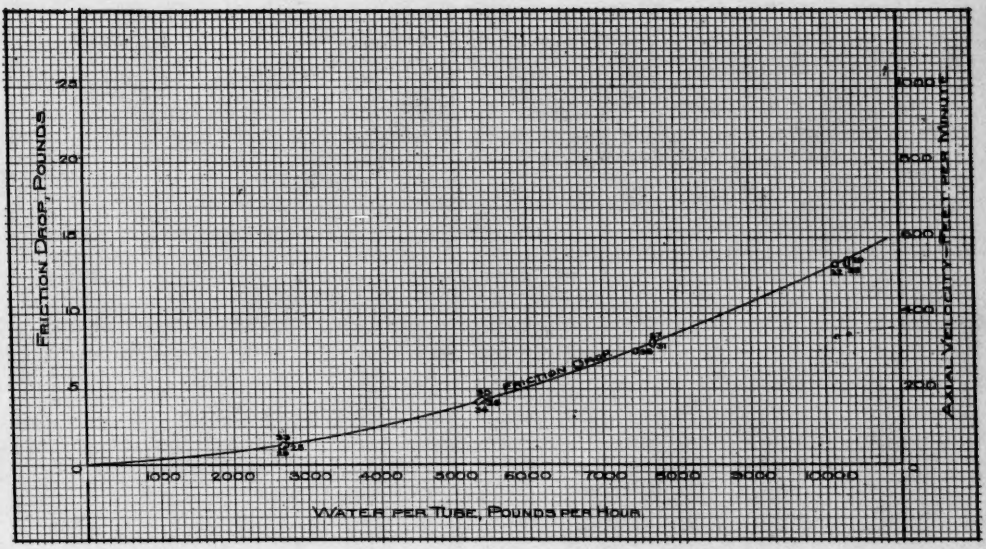


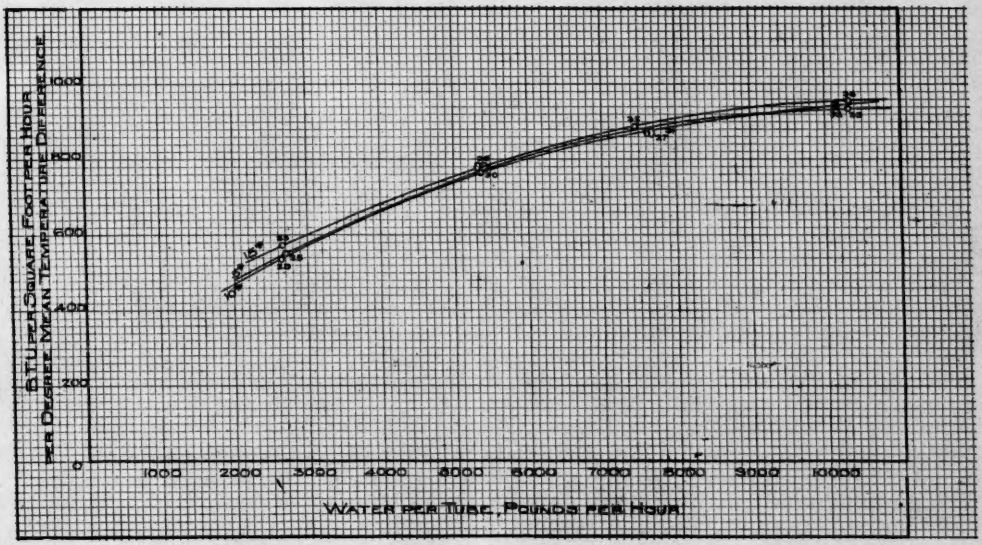


CURVES FOR TEST 285.—SCHUTTE-KOERTING FEED-WATER HEATER. VERTICAL ARRANGEMENT. 3/

NOTE.—Small numerals ind

FIG. 1

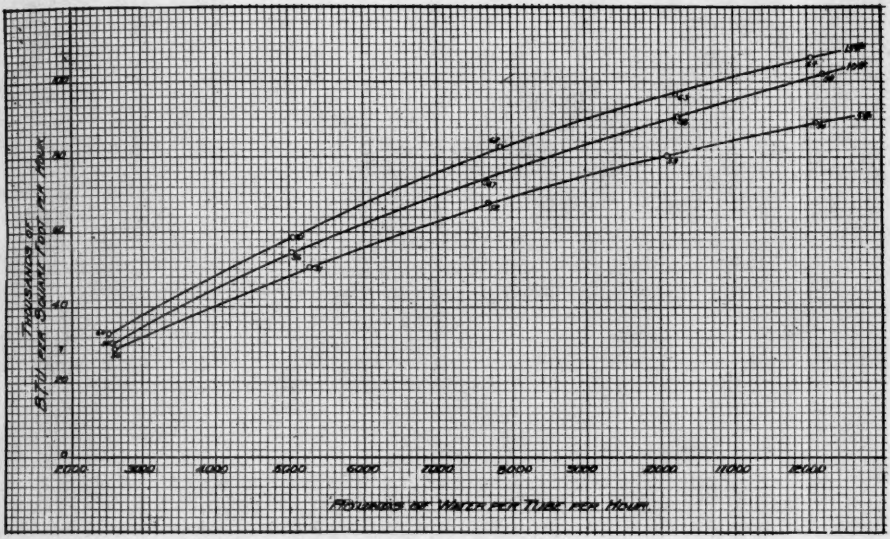


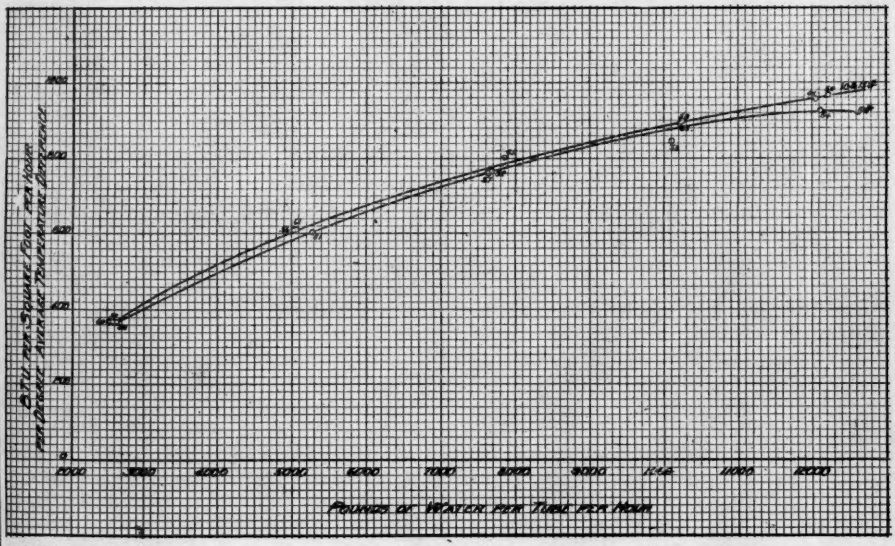


EMENT. 3/16-INCH FILM THICKNESS. U. S. NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MD.

l numerals indicate number of run.

FIG. 13.

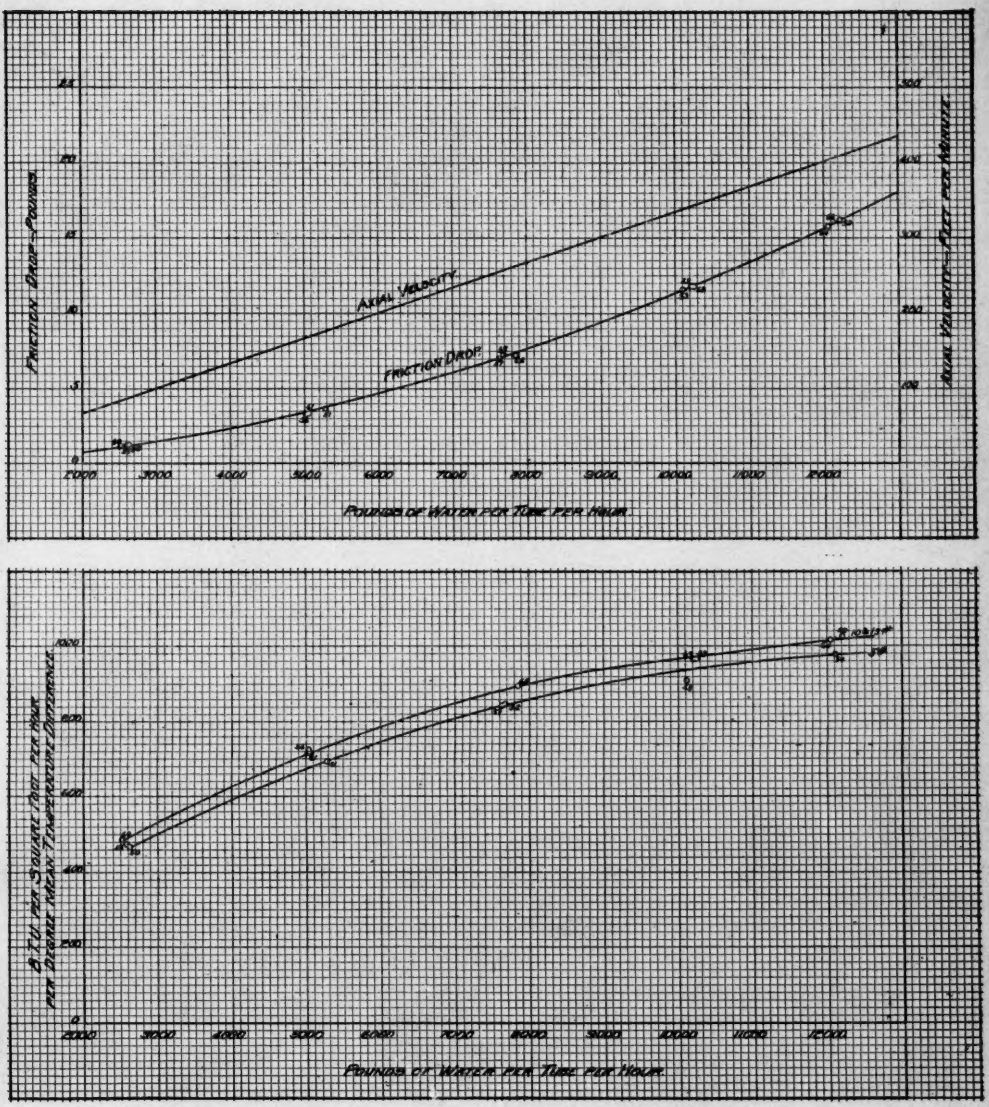




CURVES FOR TEST 285.—SCHUTTE-KOERTING FEED-WATER HEATER. HORIZONTAL ARRANGEMENT. 3/

NOTE.—Small numerals indic

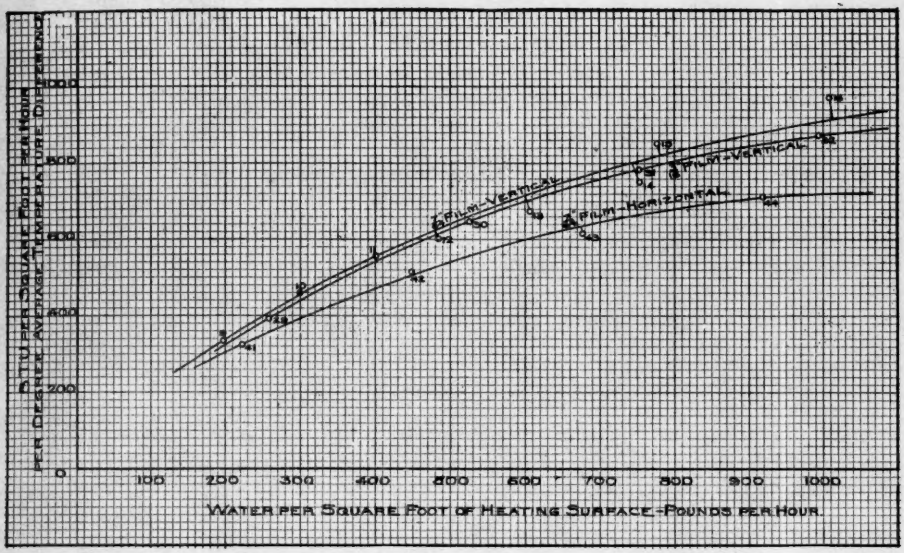
FIG. 14.

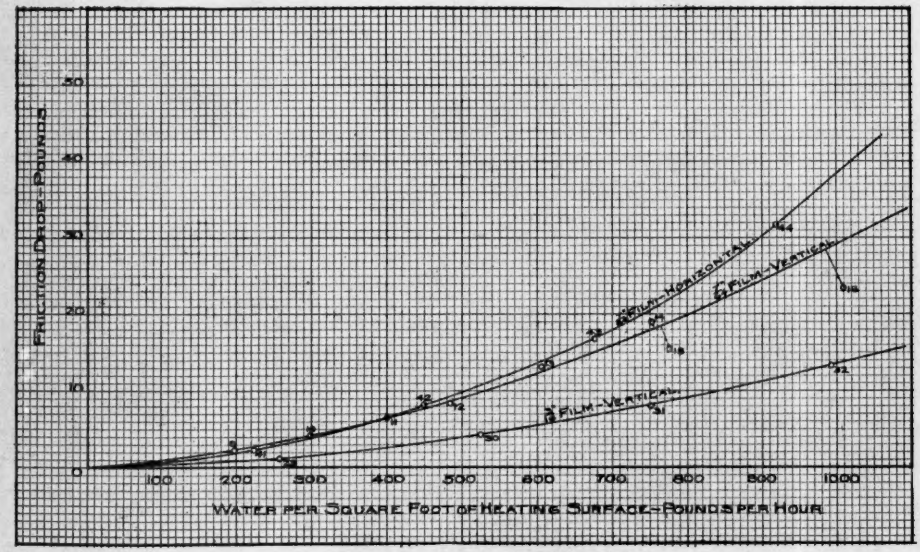

FRICTION DRAG—POUNDS
AXIAL VELOCITY—FEET PER MINUTE
AXIAL VELOCITY
FRICTION DRAG
POUNDS OF WATER PER TUBE PER HOUR
B.T.U. PER SQUARE FOOT PER HOUR
PER DEGREE MEAN TEMPERATURE DIFFERENCE
POUNDS OF WATER PER TUBE PER HOUR

EMENT. 3/16-INCH FILM THICKNESS. U. S. NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MD.

umerals indicate number of run.

FIG. 14.

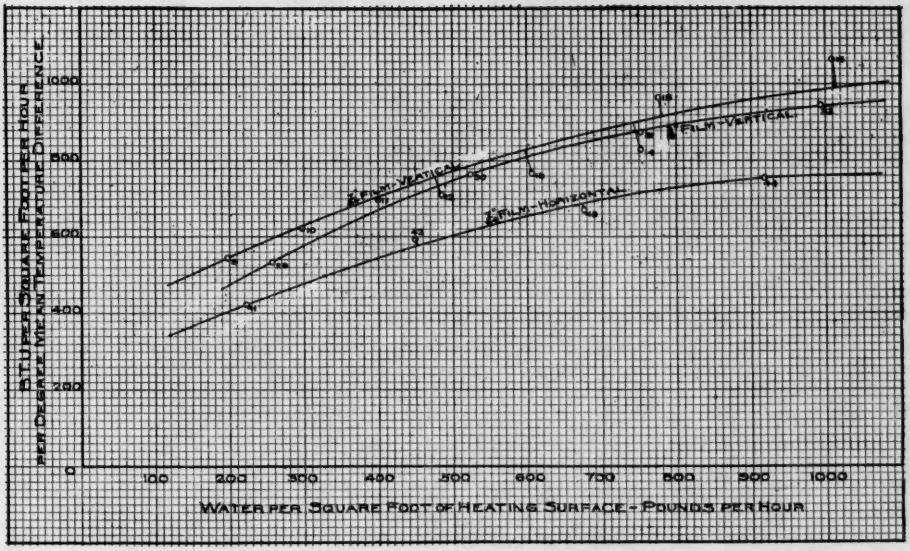


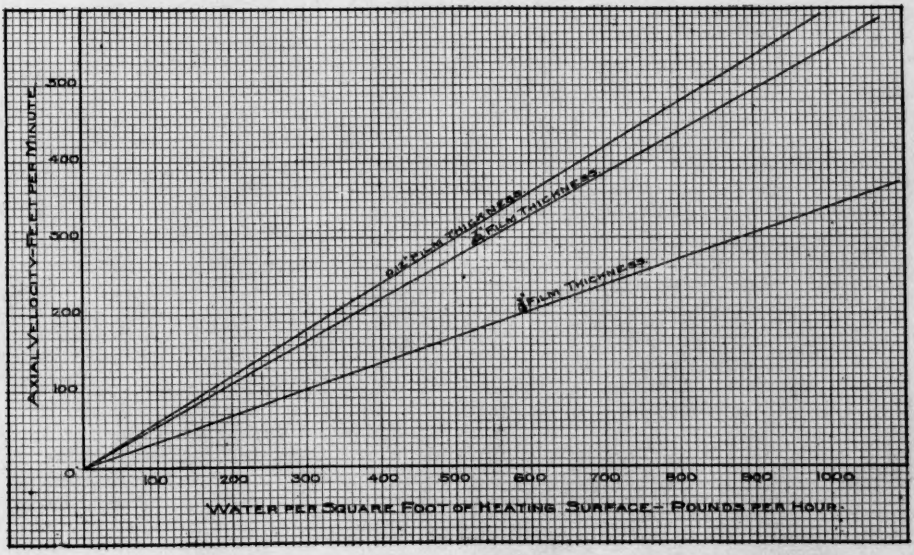


CURVES FOR TEST 285.—SCHUTTE-KOERTING FEED-WATER HEATER. EFFECT OF FILM THICKNESS

NOTE.—Small numerals indicate number of run. All runs, from which above

FIG. 15.





Thickness and Arrangement. U. S. Naval Engineering Experiment Station, Annapolis, Md.

n which above curves were plotted, made with ten pounds inlet steam pressure.

Fig. 15.

showing the characteristics in the same degree as the exponential temperature difference. These results are shown graphically in figures 11 and 12 for the 7/64-inch film, and in figures 13 and 14 for 3/16-inch film. It is to be noted that from figures 11 and 13 the rates of heat transmission are nearly the same with 7/64-inch as with 3/16-inch films at the same rate of flow per tube. This suggested the plan of plotting the results as a function of the pounds of water per square foot per hour, since the heating surface for the 3/16-inch film is slightly less than for the 7/64-inch film. The resulting curves are presented in figure 15 for a steam pressure of 10 pounds gage. It appears that for the same rate of surface flow—pounds of water per square foot of heating surface per hour—the 7/64-inch film shows from 3½ to 4¼ per cent. greater heat transmission for equal rates of flow than the 3/16-inch film. It accomplishes this increase, however, with a friction loss two and one-half times as great.

Considering now the relation between vertical and horizontal heaters of the same film thickness: At 500 pounds of water per unit of surface the horizontal heater shows rates of heat transmission of 720 B.t.u. per hour per degree temperature difference as against 910 B.t.u. for the vertical heater, an advantage of 26.4 per cent. in favor of vertical installation. Care was taken to give the horizontal heater sufficient pitch to cause the water of condensation to flow freely from the tubes. This did not entirely accomplish the desired result, as the weights of condensed steam for the several intervals of any one test varied between wide limits, whereas in the vertical heater these quantities remained remarkably constant. However, the great difference between the performance of the horizontal and vertical heater cannot be entirely accounted for by the accumulation of water of condensation in the lower part of the corrugations of the inner tube, such as would render partly inactive less than 5 per cent. of the total surface. A reason is, however, to be found in the accumulation of air in the steam space. Air at a given pressure has a greater density than steam at the same pressure and corresponding tempera-

ture of saturation. Hence the air will accumulate over collected water, and the usual precautions of blowing through pet cocks in the covers would be ineffective with such an installation. The presence of this air would actively reduce the transmitting surface, and this, taken with the irregular flow of water of condensation, would explain the variation. In a subsequent test a more efficient disposal of air, attendant, however, with considerable loss of steam, resulted in rates of heat transmission of about 5 per cent. less for horizontal than for vertical installation.

There is a noticeable lack of uniformity between the friction losses for high and low rates of flow with 7/64-inch-film vertical heater, and between the results with vertical and horizontal heater, 7/64-inch film. With a small film the clearances cannot be considered to remain fixed for any period of time. Friction is altered by simply turning steam on a cold heater, hence the clearances and adjustments of the tubes are altered materially by the action of heat as well as by possible interchanges of units when the apparatus is assembled after cleaning.

The 3/16-inch film, on the other hand, gives uniform values of friction loss under widely varying temperature limits, and possesses the further great advantage of replacement of tubes without danger of change in adjustment sufficient to modify the nominal film thickness. The greater uniformity in heat transfers with this film over the 7/64-inch film is likely due to the same cause. Uniform film means uniform agitation and ability to reproduce results, whereas variations in clearance produce temporary obstructions which increase agitation and hence modify heat transfer.

The standardization of the 3/16-inch-film vertical heater is therefore advisable from the important considerations of minimum friction loss, equal heat transfer per unit of heating surface, efficient removal of water of condensation and non-condensible gases from contact with the heating surface and the ability to maintain uniform film thickness when assembling the units.

VI.—Design Curves for Feed Heaters.

The primary object of such tests as have been cited is to obtain data of value in design. Following the suggestions of the Bureau of Steam Engineering these data have been incorporated in a number of design curves which permit a graphical solution of all problems of feed-heater proportions for a given type of heater.

In any case the quantity of feed to be heated, the amount of auxiliary steam and its pressure and quality and the hotwell temperature will be known, and hence the problem resolves itself to finding the heater best suited to the conditions together with the final feed temperature resulting from the design. Obviously the most efficient heater is one which absorbs all of the latent heat in the auxiliary exhaust and sends it as water to the hotwell without permitting the auxiliary condenser to extract useful feed heat. Hence the available latent heat in auxiliary steam can be entirely converted into sensible heat of feed. As a rule auxiliary exhaust will carry about 15 per cent. moisture leaving 85 per cent. as dry steam. The relation between heat in steam and in feed will be given by

$$.85\ S \times r = W\ (t_o - t_i),$$

where

S = weight of auxiliary steam;
r = heat of vaporization of steam at auxiliary exhaust pressure;
W = weight of feed, all purposes;
t_o = temperature of feed leaving heater;
t_i = temperature of feed entering heater.

For convenience in calculations W is given as thousands of pounds of feed, whence S is pounds of auxiliary steam per 1,000 pounds of feed, and the above relation may be expressed

$$\frac{850\ Sr}{W} = (t_o - t_i).$$

Above a minimum value of t_i, which in practice will be 75 degrees F., the dry auxiliary steam per 1,000 pounds of feed

will be directly proportional to temperature, t_o, as shown by the straight line on figure 16, where dry steam per 1,000 pounds of feed is plotted as the ordinate with temperature as the corresponding abscissae. The second curve, length of tube against temperature of feed, is the temperature gradient as derived from experiment according to the tests described in section IV. This curve gives for one rate of flow and for the heater in question the temperature at any distance along the equivalent straight tube when the inlet water temperature has the minimum value of 75 degrees. The final curve, friction loss plotted against length of tube, measures the drop corresponding to any one rate of flow. The term "unit tube" is used because the unit is taken as the length of tube in one pass of the heater tested.

Thus there result three curves for each rate of flow and each steam pressure giving all the heaters which will absorb the entire latent heat in the auxiliary steam available. Figures 16 and 17 give the data of Tables IV and V in the form of design curves for the Bureau heater with and without retarders. To illustrate the use of these curves let it be required to design one of two feed heaters of the Bureau type without retarders for a ship using 420,000 pounds of steam per hour, all purposes, of which 12 per cent. is auxiliary steam at 10 pounds pressure.

Dry auxiliary steam per hour $= .12 \times .85 \times 420{,}000 =$ 42,840 pounds.

Dry auxiliary steam per heater per hour $=$ 21,420 pounds.

Main feed per heater per hour $=$ 210,000.

$$\text{Dry steam per 1,000 pounds of feed} = \frac{21{,}420}{210} = 102.$$

Hotwell temperature $=$ 92 degrees.

Feed per tube per hour $=$ 3,000 pounds.

$$\text{Number of tubes required} = \frac{210{,}000}{3{,}000} = 70.$$

Since the curves are based on a minimum temperature of 75 degrees, it is necessary to get the following data corresponding to 92 degrees:

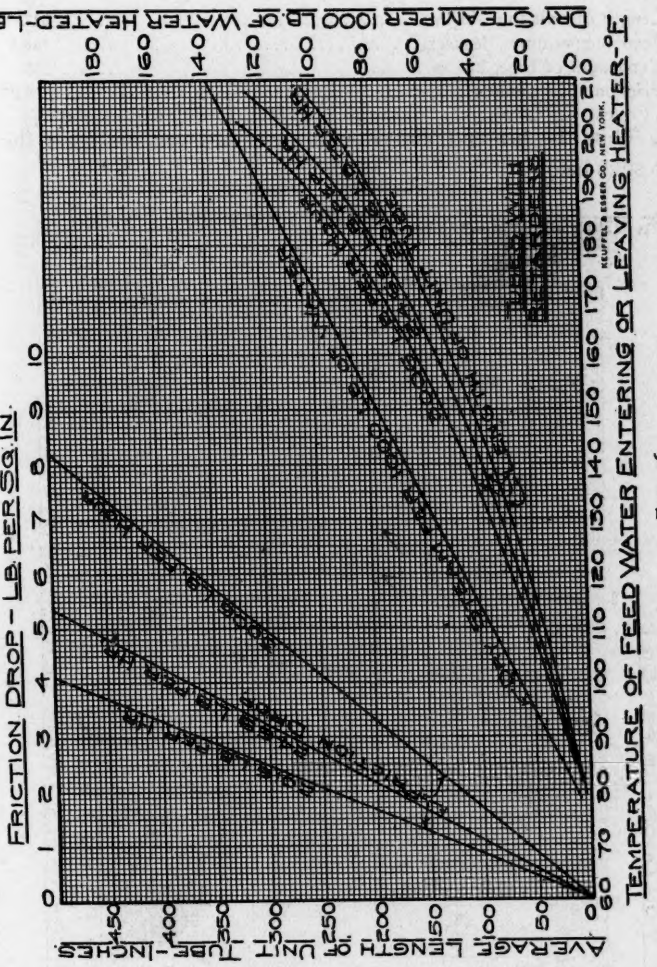

AVERAGE LENGTH OF UNIT TUBE-INCHES
FRICTION DROP - LB. PER SQ. IN.
TEMPERATURE OF FEED WATER ENTERING OR LEAVING HEATER-°F
DRY STEAM PER 1000 LB. OF WATER HEATED-LB
KEUFFEL & ESSER CO., NEW YORK.

FIG. 16.

Length of unit tube, inches..... 32
Dry steam per 1,000 pounds feed, pounds..... 18

Hence

Total dry steam per 1,000 pounds of feed = 18 + 102 = 120.

Corresponding to 120 pounds of dry steam the curve gives:

Length of unit tube, inches	317.
Feed temperature, degrees	186.5
Net length of tube, inches	285.
Friction loss, pounds	5.27

Thus the heater obtained will have the sizes and give the results below:

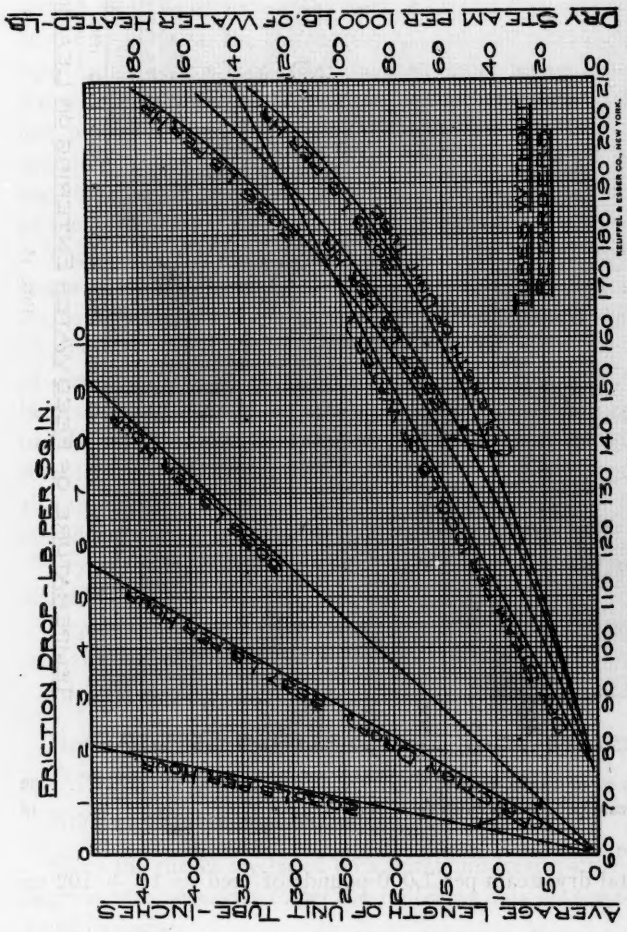

FRICTION DROP - LB. PER SQ. IN.
AVERAGE LENGTH OF UNIT TUBE - INCHES
TEMPERATURE OF FEED WATER ENTERING OR LEAVING HEATER - °F.
DRY STEAM PER 1000 LB. OF WATER HEATED - LB.
KEUFFEL & ESSER CO., NEW YORK.

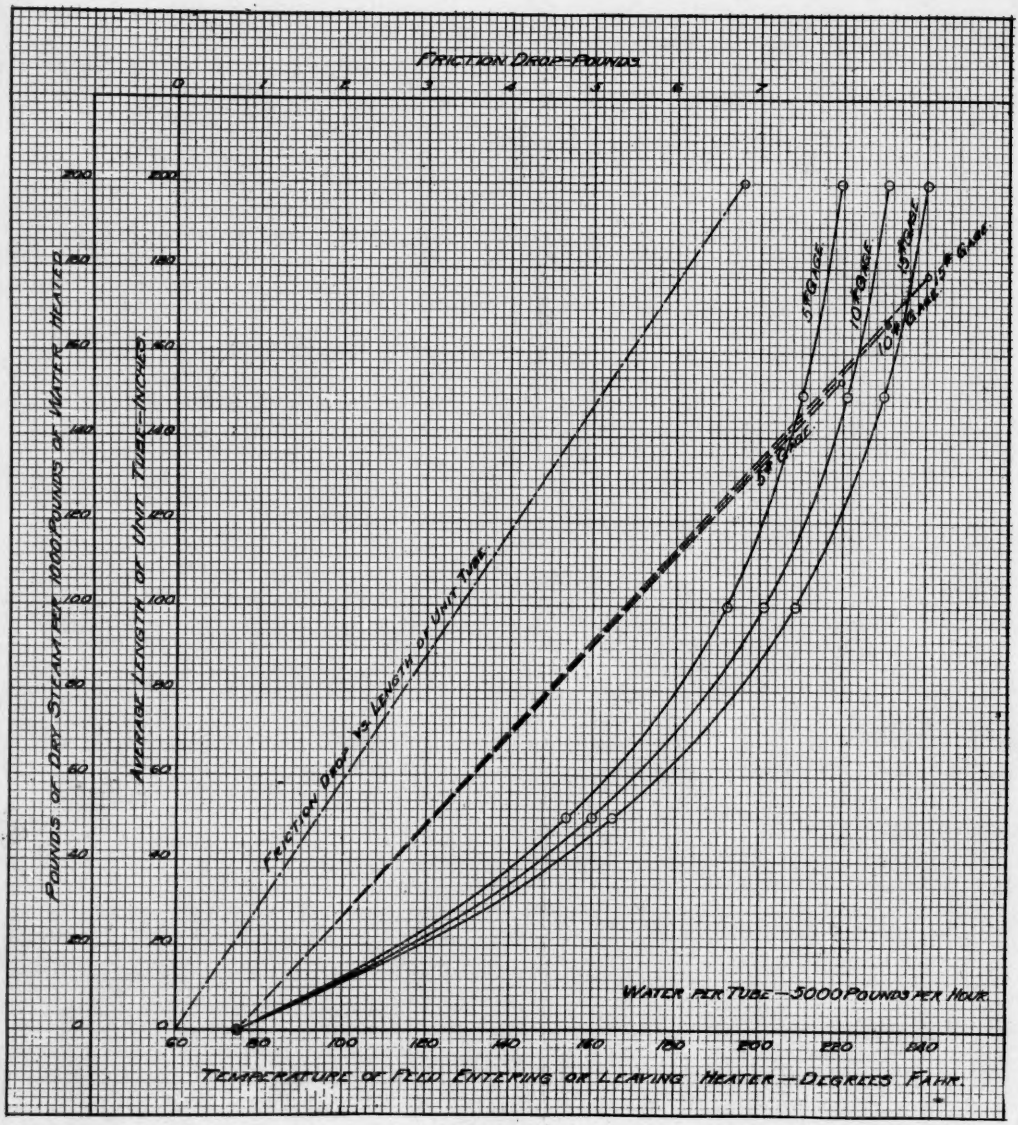
FIG. 17.

. 317.
. 186.5
. 285.
5.27

ve the

WATER ENTERING OR LEAVING HEATER-°F

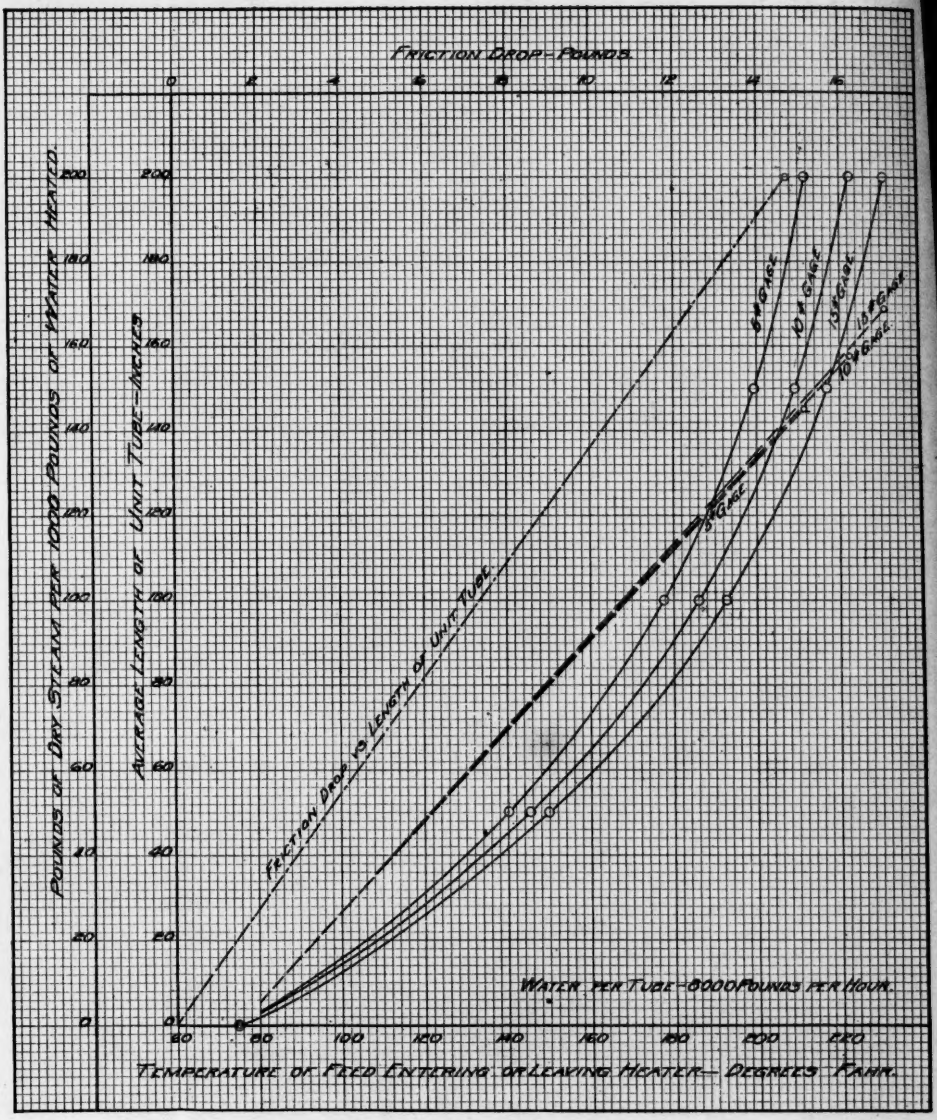
FIG. 17.


Friction Drop-Pounds
0 1 2 3 4 5 6 7
Pounds of Dry Steam per 1000 Pounds of Water Heated
Average Length of Unit Tube-Inches
Friction Drop vs Length of Unit Tube
5 Stage
10 Stage
5 Stage
10 Stage
Water per Tube—5000 Pounds per Hour
60 80 100 120 140 160 180 200 220 240
Temperature of Feed Entering or Leaving Heater—Degrees Fahr.

DESIGN CURVES, SHOWING EFFECT OF VARYING STEAM PRESSURE.——SCHUTTE-KOERTING FEED-WATER HEATER WI

——————— Average Length of Unit Tube. —— — — — P

FIG. 18.



HEATER WITH 3/16-INCH FILM.——U. S. NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MD.

—— —— Pounds of Dry Steam per 1,000 Pounds of Water Heated.

Number of tubes per pass	70.
Total length of unit tube, inches	285.
*Length of unit tube per pass, inches	47.5
Number of passes	6.
Inlet-water temperature, degrees	92.
Final feed temperature, degrees	186.5
Friction loss, pounds	5.27
Heating surface per heater, square feet	334.

*Assumed as unit nearest 45.

This is a practical heater, inasmuch as the friction loss is less than 10 pounds for the maximum rate of flow. At rates of flow corresponding to 1,500, 2,000 and 2,500 pounds per tube, or 210,000, 280,000 and 350,000 pounds of feed, all purposes, the feed temperature, if the exhaust pressure remain constant, at 10 pounds will be 210 degrees, 201.5 degrees, and 190 degrees respectively.

Based on the data of Tables IX, X and XI, design curves for the Schutte and Koerting feed heater have been derived and are presented in figures 18, 19 and 20 for water rates of 5,000, 8,000 and 10,000 pounds per tube per hour and steam pressures of 5, 10 and 15 pounds. These curves are likewise based on a minimum hotwell temperature of 75 degrees F. A water rate of 5,000 pounds per tube is the equivalent of builder's rated capacity of the heater tested, but the test results indicated that full economy of heating surface was not realized at that rate. The higher values of 8,000 and 10,000 pounds were selected because they fall about evenly on the two sides of the present allowable friction drop of 10 pounds for a two-pass heater. The mean length of tube in each of two passes is 51⅝ inches, and the heating surface for a pair of tubes is 5.11 square feet; so the equivalent heating surface for lengths of 50, 100, 150 and 200 inches would be 4.95, 9.90, 14.85 and 19.80 square feet.

The design curves in figures 18 and 19 show the variation of outlet temperature, dry steam per thousand pounds of water and friction drop with length of tube and with steam pressure when the rate of flow in pounds per tube is maintained constant, whereas the curves in figure 20 show the variation of

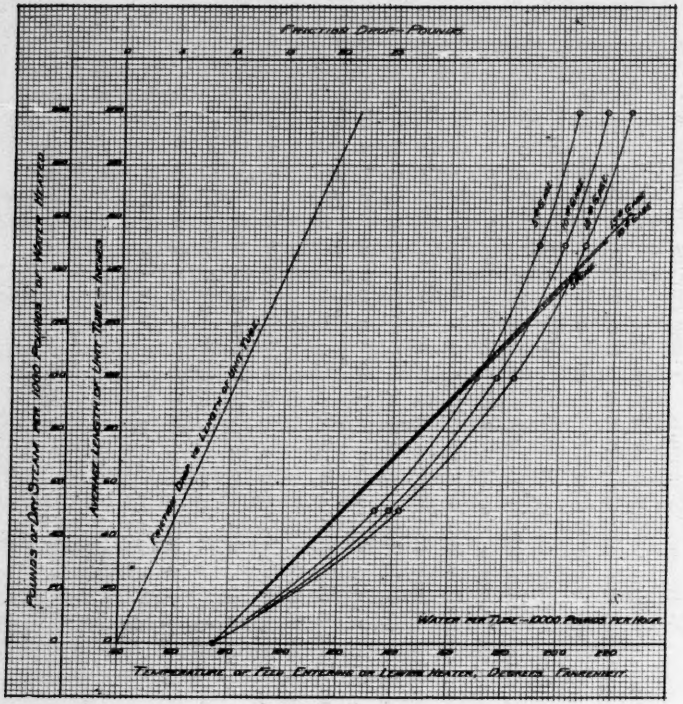

Friction Drop - Pounds
Pounds of Dry Steam per 1000 Pounds of Water Heated
Average Length of Unit Tube - Inches
Friction Drop vs Length of Unit Tube
Water per Tube - 1000 Pounds per Hour
Temperature of Feed Entering or Leaving Heater, Degrees Fahrenheit

FIG. 19.—DESIGN CURVES SHOWING EFFECT OF VARYING STEAM PRESSURE. SCHUTTE-KOERTING FEED-WATER HEATER WITH 3/16-INCH FILM. U. S. NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MD.

Average length of unit tube.
Pounds of dry steam per 1,000 pounds of water heated.

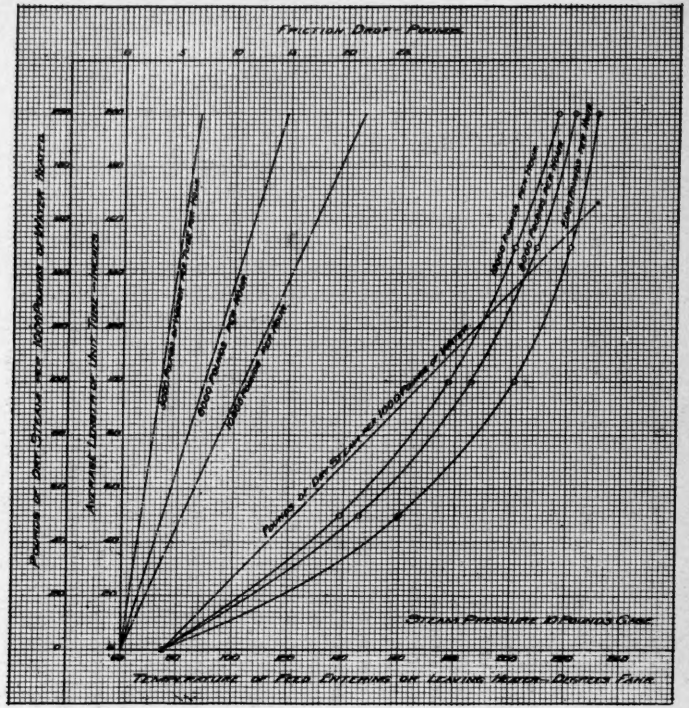

Friction Drop - Pounds
Pounds of Dry Steam per 1000 Pounds of Water Heated
Average Length of Unit Tube - Inches
Steam Pressure 10 Pounds Gage
Temperature of Feed Entering or Leaving Heater, Degrees Fahr.

FIG. 20.—DESIGN CURVES SHOWING EFFECT OF VARYING WATER RATE. SCHUTTE-KOERTING FEED-WATER HEATER WITH 3/16-INCH FILM. U. S. NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MD.

Average length of unit tube.
Pounds of dry steam per 1,000 pounds of water heated.
Friction drop *vs.* length of unit tube.

the several quantities with rate of flow when steam pressure is maintained constant at 10 pounds gage.

VII.—Effect of Operating Conditions on Feed Temperature.

At a water rate of 8,000 pounds per tube per hour the length of tube required to heat water from 90 degrees to 210 degrees with exhaust steam at 10 pounds gage is 144 inches. Reducing the steam pressure to 5 pounds with the same tube would yield an outlet temperature of 200 degrees, whereas increasing the pressure to 15 pounds would increase the outlet to 218 degrees F. A decrease of 10 degrees in feed temperature represents a loss of 10 B.t.u. per pound of steam generated, or 1 per cent. of the heat absorbed in the boilers, whereas decreasing back pressure from 10 to 5 pounds would effect a reduction in steam consumption of turbine-driven auxiliaries from 43.9 to 40.2 pounds per shaft horsepower per hour,* a saving of 8½ per cent. in this item of auxiliary steam. Thus where auxiliary steam is in excess of 12 per cent. of total steam it appears to be in the interest of overall economy to obtain as low an auxiliary exhaust pressure as possible and sacrifice high feed temperature.

Considering the effect of variation in water rate at a constant steam pressure, the curves show the length of tube required to heat water from 90 degrees to 210 degrees at 10,000 pounds per tube to be 166 inches with a friction drop of 17.9 pounds. If the heater be designed for these conditions for maximum load, then the outlet temperatures for 8,000 and 5,000 pounds per tube would be 217 degrees and 226 degrees with corresponding friction drops of 12.2 and 5.7 pounds.

The economy of increased water rate may best be shown by calculations for one of two heaters for a ship whose steam consumption for all purposes is 480,000 pounds per hour of

*See report of test of Worthington centrifugal feed pump driven by Terry turbine, Journal American Society of Naval Engineers. Assumed mechanical efficiency of pumps is 50 per cent.

U. S. Naval Engineering Experiment Station, Annapolis, Md.
Average length of unit tube.
Pounds of dry steam per 1,000 pounds of water heated.
Friction drop vs. length of unit tube.
U. S. Naval Engineering Experiment Station, Annapolis, Md.
Average length of unit tube.
Pounds of dry steam per 1,000 pounds of water heated.

which 60,000 pounds is used by auxiliaries. The results are as follows:

Total water fed, pounds per hour			480,000.
Auxiliary steam, pounds per hour			60,000.
Quality of auxiliary exhaust			.85
Water per heater, pounds per hour			240,000.
Dry auxiliary steam per heater, pounds per hour			25,500.
Temperature of inlet to heater, degrees			90.
Water, per tube	5,000	8,000	10,000
Number of tubes	48	30	24
Dry steam per 1,000 lbs. of feed per hour	106	106	106
Add for increased temperature of inlet water	16	16	16
Total steam per 1,000 lbs. of feed	122	122	122
Corresponding outlet temperature	189	189	189
Length of tube, inches	82	107	122
Subtract for 90 degrees inlet water	6	9	10
Net length of tube, inches	76	98	112
Friction drop, pounds	2.6	7.2	11.0
Heating surface per inch, square foot	.099	.099	.099
Total heating surface per unit	361.	291.	266.

Selecting the 112-inch heater with 24 units would produce results at 50 per cent. and 80 per cent. capacity of

Water, per tube, pounds	5,000.	8,000.
Water, per heater, pounds	120,000.	192,000.
Outlet feed temperature with 10 lbs. auxiliary exhaust	211.5	197.
Outlet feed temperature with 5 lbs. auxiliary exhaust	201.5	189.
Friction drop, pounds	3.5	7.5

In each case at reduced power there would be excess auxiliary steam to be bled to the auxiliary condenser, whereas at full power the heat absorbed by the feed heaters should care for the entire condensation.

VIII.—Air in Feed Heaters.

In the tests previously noted precautions were taken to vent all accumulated air from the steam side of the heaters. Water will absorb nearly 2 per cent. by volume of air and this air will

be carried over by the steam. As has been pointed out, at a given pressure and temperature of saturation air has a greater density than the steam with which it is associated, and it will gradually accumulate above the condensation in the bottom of the heater. Thus part of the surface is rendered inactive and the film resistance is increased to an extent that heat transfer is appreciably reduced. This blanketing effect will be greater in a film heater with horizontal tubes than if the tubes be vertical, and in the latter case the usual precautions of blowing through pet cocks in the covers would be ineffective.

The extent to which accumulated air will reduce heat transfer is to be noted from Table XII, which contains data from a two and one-half hour test on the film heater, during which time no air was relieved from the steam space and the water seal permitted only the release of water. The rate of heat transmission decreased steadily from 661 B.t.u. per square foot per degree average temperature difference to 558 B.t.u., a net loss of 15.6 per cent. At the same time the hot feed temperature fell from 195.1 degrees to 181.8 degrees. Although full steam pressure of 10 pounds was maintained, the temperature at the bottom of the heater fell from 234.7 degrees to 216.5 degrees, and two lower temperatures were noted during the test. Automatic air valves are not suitable for this service, and the best remedy is pet-cock relief with a moderate but continuous blow of steam from the bottom of the steam space.

IX.—Auxiliary Steam for Feed Heating.

The amount of auxiliary steam available for feed heating depends on the type and size of vessel and the character of the main and auxiliary equipment. From the trial trips which have been reported recently in the Journal of the American Society of Naval Engineers the data of Table XIII has been collected, incorporating the steam consumption of such vessels where the main-engine steam was separated from the auxiliary steam.

				Water Data.									Steam Data.							Deduced Data.				
Number of Observation.	Date of Run. October, 1912.	Time of Observation.	Room Temperature.	River Water Pumped Through Heater. Pounds per Hour.	Temperature at Inlet. Degrees F.	Temperature at Outlet. Degrees F.	Temperature Rise. Degrees F.	Specific Heat of Water.	B.T.U. Absorbed by Water per Hour.	Friction Drop. Inches of Mercury.	Water Pressure. Outlet.	Friction Drop. Pounds per Sq. In.	Pressure by Gage at Inlet.	Pressure. Pounds Absolute. at Inlet.	Temperature at Inlet. Degrees F.	Temperature corresponding to Pressure.	Degrees Superheat.	Temperature at Outlet of Heater. Degrees F.	Temperature of Condensed Steam.	B.T.U. Transmitted per Sq.Ft. per Hour.	Average Arithmetical Temperature Difference.	B.T.U. Absorbed per Sq.Ft. per Hour per Degree Average Temp. Diff.	Water Heated per Sq.Ft. of Heating Surface. Pounds per Hour.	Water per Tube. Pounds per Hour.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
1	25	1:30	75	——	61.9	195.1	133.2	.974	------	24.0	183	10.9	10.3	25.0	301.6	240.1	61.5	234.7	234.2	-----	111.6	---	---	----
2	25	1:40	76	12151	61.9	194.4	132.5	.974	1568700	22.3	183	10.2	10.0	24.7	293.8	239.4	54.4	231.8	230.9	73540	111.2	661	570	6076
3	25	1:50	77	12112	61.8	193.0	131.2	.974	1547900	21.8	187	9.9	9.8	24.5	291.0	239.0	52.0	228.5	228.1	72970	111.6	650	568	6056
4	25	2:00	75	11719	61.8	189.7	127.9	.974	1460200	21.0	183	9.6	9.9	24.6	287.9	239.2	48.7	228.0	225.3	68450	113.4	604	549	5860
5	25	2:10	72	11719	61.8	194.2	132.4	.974	1511800	21.0	188	9.6	10.0	24.7	273.5	239.4	34.1	225.4	224.7	70870	111.4	636	549	5860
6	25	2:20	73	11680	61.9	191.7	129.8	.974	1476400	20.0	185	9.1	10.0	24.7	266.4	239.4	27.0	222.3	222.1	69210	112.6	615	548	5840
7	25	2:30	73	11837	62.0	199.0	137.0	.974	1579100	22.8	180	10.4	10.0	24.7	266.1	239.4	26.7	222.7	222.3	74030	108.9	680	555	5919
8	25	2:50	73	13095	61.9	192.4	130.5	.974	1664400	24.8	229	11.3	10.0	24.7	266.5	239.4	27.1	218.7	219.9	78030	112.2	695	614	6548
9	25	3:00	74	12899	62.0	192.9	130.9	.974	1644600	22.1	226	10.1	10.0	24.7	272.0	239.4	32.6	221.4	220.1	77100	111.9	689	605	6450
10	25	3:10	73	11798	62.1	189.1	127.0	.974	1459400	22.4	236	10.2	10.0	24.7	267.6	239.4	28.2	220.0	218.9	68420	113.8	601	553	5899
11	25	3:20	73	11994	62.1	189.9	127.8	.974	1493300	22.2	236	10.1	10.0	24.7	267.9	239.4	28.5	218.5	217.9	70000	113.4	617	562	5997
12	25	3:30	74	11915	62.2	189.3	127.1	.974	1475100	21.8	238	9.9	9.9	24.6	269.2	239.2	30.0	217.7	216.9	69150	113.4	610	559	5958
13	25	3:40	74	11798	62.2	189.9	127.7	.974	1467700	21.6	236	9.8	10.1	24.8	264.1	239.6	24.5	212.5	212.3	68810	113.5	606	553	5899
14	25	3:50	74	11758	62.2	186.2	124.0	.974	1420400	21.5	231	9.8	10.0	24.7	270.4	239.4	31.0	207.7	208.3	66990	115.2	578	551	5879
15	25	4:00	74	11994	62.2	181.8	119.6	.974	1397300	22.4	231	10.2	10.0	24.7	241.4	239.4	2.0	216.5	216.1	65510	117.4	558	562	5997

TABLE XII.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER. 7/64-INCH WATER FILM.—VERTICAL ARRANGEMENT.

TABLE XIII.—MAIN AND AUXILIARY STEAM CONSUMPTION OF NAVAL VESSELS.

Vessel and type.	Speed, knots.	Steam per hour, pounds.		Auxiliary steam, per cent. of total.	Auxiliary exhaust pressure.	Feed temperature.	
		Auxiliaries.	All purposes.			At hot well.	At boilers.
Delaware	21.56	40,557	422,931	9.6	...	104.5	....
Engined battleship.	19.22	30,512	243,810	12.5	...	89.7	....
	12.24	23,457	83,462	28.1	...	90.7	....
Utah	21.04	62,400	406,397	15.4	...	90.1	173.1
Turbined battleship.	19.23	47,400	262,498	18.1	...	91.5	177.5
	12.02	32,200	98,498	32.7	...	93.0	213.0
Florida	22.08	67,352	560,880	12.0	6.7	74.0	197.8
Turbined battleship.	19.19	56,382	308,051	18.0	4.9	81.9	201.0
	12.08	41,843	116,386	35.9	6.4	79.5	223.3
Birmingham	10.03	12,081	34,069	35.2	...	....	....
Engined scout cruiser.	14.95	15,657	65,412	33.9	...	60.0	146.4
	19.86	22,217	132,037	16.8	...	...	...
	22.55	29,684	214,894	13.8	...	....	....
	24.15	43,928	297,300	14.8	...	....	....
Salem	10.20	13,446	46,598	28.8	...	....	....
Turbined scout cruiser.	14.85	17,967	82,947	21.7	...	73.5	172.9
	20.16	20,337	157,519	12.9	...	...	...
	22.55	33,520	224,826	14.9	...	...	...
Chester	10.15	14,151	47,690	29.6	...	...	...
Turbined scout cruiser.	15.81	18,369	95,621	19.2	...	98.0	212.2
	20.05	25,569	158,220	16.1	...	...	....
	22.78	29,002	205,553	14.1	...	...	...
	24.67	42,392	296,054	14.3	...	...	...
Parker and *Benham*	29.57	36,146	283,633	12.7	...	56.9—59.4	183.3—193.3
Destroyers, turbines and	24.06	18,066	138,491	13.0	...	53.3—55.3	189.6—209.3
cruising engines.	15.56	5,523	32,281	17.1	...	46.1—51.8	210.3—214.4
	12.00	5,253	19,717	26.6	...	46.0—53.6	199.5—221.6
Paulding	32.80	33,000	218,655	15.1	...	96.4	187.2
Turbined destroyer.	25.31	23,000	105,630	21.8	...	88.0	221.0
	15.86	12,500	30,967	40.4	...	71.0	222.0
New York	21.47	52,336	495,183	10.56	5.3	94.0	190.6
Engined battleship.	19.23	53,296	300,038	17.78	11.8	92.8	216.8
	12.11	36,236	102,645	35.42	14.9	76.7	236.2

Where steam-consumption guarantees of machinery for proposed installations are not available, this table may with safety be used for a vessel of similar class and equipment.

X.—CONCLUSIONS.

From the tests considered herein the following summary applies:

(1) Closed feed-water heaters developed for marine purposes have temperature gradients resulting from the general

relation that heat transfer is proportional to some power of the temperature difference;

(2) The numerical value of the exponent for the heaters tested is less than unity;

(3) A vertical tube permits higher rates of heat transfer than a horizontal tube due to the more effective disposal of condensate;

(4) Film-tube heaters give much higher rates of heat transfer than plain tubes or tubes with retarders;

(5) The clearance between the tubes making up a film unit can be great enough to prevent excessive friction loss in the form of pressure drop without in the slightest decreasing the efficiency of the film agitation, which renders this type more effective for heat transfer than a plain tube;

(6) Air carried into the steam space of a feed-water heater will seriously interfere with heat transmission in any type of heater if allowed to accumulate for any considerable period of time;

(7) Water-film agitation is more important than steam side-film agitation;

(8) Water-film agitation and air-free condensation are essential to the maintenance of high rates of heat transfer.

APPENDIX I.

Principal Dimensions of Schutte and Koerting Film-Oil Heater Illustrated in Figure 8.

The following dimensions were obtained by measurement after the completion of the test:

Weight, empty, pounds	507½
Length over all, feet and inches	6–05
Width over external steam connections, inches	24
External diameter, inches	11⅛
Outer tube:	
Inside diameter, inches	4.03
Depth of corrugations, inch	0.62
Pitch of corrugations, inches	2
Thickness (assumed), B.W.G	9
Length between headers, inches	43½

Inner tube:	
Outside diameter, inches	5.04
Depth of corrugations, inch	0.68
Thickness, inch	0.11
Length between heads, inches	54
Steam connections, inch	1
Oil connections, inches	1½
Distance between center lines of oil connections, inches	41½
The following values have been calculated from the above dimensions:	
Outer surface of outer tube, square feet	5.58
Inner surface of inner tube, square feet	6.13
Total heating surface, square feet	11.71
Maximum inside diameter of outer tube, inches	5.27
Maximum outside diameter of inner tube, inches	5.04
Clearance, inch	0.115
Sectional area of water film, square inches	1.86
Minimum inside diameter of outer tube, inches	4.03
Minimum outside diameter of inner tube, inches	3.68
Clearance, inch	0.175
Sectional area of water film, square inches	2.12
Average sectional area of water film, square inches	1.99

—

APPENDIX II.

Principal Dimensions of Schutte and Koerting Experimental Feed-Water Heater Illustrated in Plate II.

The principal dimensions of the apparatus are:	
Length over all, feet and inches	6–09¼
External diameter of shell, inches	10¾
Internal diameter of shell, inches	10¼
Face to face of water connections, inches	18¼
Outer tubes:	
Length over all, feet and inches	3–11
Maximum inside diameter, inches	2.373
Minimum inside diameter, inches	1.813
Maximum outside diameter, inches	2.563
Minimum outside diameter, inches	2.003
Depth of corrugations, inch	.280
Pitch of corrugations, inches	1.125
Thickness, B.W.G. number	13
Thickness, inch	.095
Inner tubes, 7/64-inch film:	
Length over all, feet and inches	4–10¼
Maximum inside diameter, inches	1.997
Minimum inside diameter, inches	1.375
Maximum outside diameter, inches	2.163
Minimum outside diameter, inches	1.541

Inner tubes, 7/64-inch film:

Depth of corrugations, inch	.311
Pitch of corrugations, inches	1.125
Thickness, B.W.G., number	14
Thickness, inch	.083

The 3/16-inch film is constructed by using the same outer tube as in the 7/64-inch film, and making the inner tube from the same blank as the inner tube with smaller film, but driving the forming tool into the blank tube to a greater depth.

The resulting dimensions for the inner tubes for 3/16-inch film would be:

Length over all, feet and inches	4–10¼
Maximum inside diameter, inches	1.936
Minimum inside diameter, inches	1.193
Maximum outside diameter, inches	2.102
Minimum outside diameter, inches	1.359
Depth of corrugations, inch	.311
Pitch of corrugations, inches	1.125
Thickness, B.W.G. number	14
Thickness, inch	.083

From the dimensions given in the first paragraph and radii of arcs of corrugations shown on Plate II the following values have been calculated:

Nominal film thickness, inch	$0.\frac{7}{64}$
Actual film thickness, inch	0.120
Outer surface of one outer tube, square feet	5.155
Inner surface of one inner tube, square feet	5.510
Total heating surface in tubes, square feet	21.33
Sectional area of each water film, normal to axis of tube, square inch	0.744
Total area of water flow, normal to axis of tube, square inches	1.488

Similarly there have been derived for the 3/16-inch film the following:

Nominal film thickness, inch	$0.\frac{3}{16}$
Actual film thickness, inch	0.1875
Outer surface of one outer tube, square feet	5.155
Inner surface of one inner tube, square feet	5.070
Total heating surface in tubes, square feet	20.45
Sectional area of each water film, normal to axis of tube, square inches	1.145
Total area of water flow, normal to axis of tube, square inches	2.290

APPENDIX III.

Method of Obtaining Items in Tables VII to XI. Test of Schutte and Koerting Feed-Water Heater.

Item,

1, 2. For reference.

3. Observed.

4. From data corrected for temperature according to Smithsonian Meteorological Tables.

5. Observed.
6. From data obtained from calibrated water meter.
7, 8. Observed.
9. Item 8 — item 7.
10. From chemist's determination.
11. From Smithsonian Physical Tables corresponding to density in item 10.
12. Item 6 × item 9 × item 11.
13. From observation of differential mercury gage.
14, 15. Observed.
16. Where friction drop was within the limit of the 40-inch differential gage used this item was calculated from item 13 × 0.4554, where 0.4554 is the decimal of a pound per square inch equivalent to 1 inch elevation of a mercury and water differential gage. Otherwise, friction drop is measured by item 14 — item 15.
17. Observed.
18. Item 17 + (0.491 × item 4).
    0.491 inches of mercury = 1 pound per square inch.
19. Observed.
20. From steam tables corresponding to item 18.
21. Item 19 — item 20.
22, 23. Observed.
24. $Q = H - q$,
    where
    Q = B.t.u. taken from each pound of steam.
    H = Heat content of one pound of steam corresponding to conditions of items 18 and 21.
    q = Heat of the liquid corresponding to item 23.
25. Observed.
26. Item 24 × item 25.
27. Item 26 — item 12.
28. From formula $W' = W \frac{H - q}{H' - q}$.
    where
    W' = Equivalent dry steam,
    W = Actual steam per hour = item 25,
    H = Total heat in one pound of inlet steam corresponding to items 18 and 21,
    H' = Total heat in one pound of dry, saturated steam corresponding to item 18,
    q = Heat of the liquid corresponding to item 23.
29. $V = \frac{W}{D \times \frac{A}{144} \times 60}$,
    where
    V = Axial velocity in feet per minute,
    W = Weight of feed water per hour = item 6,

A = Area in square inches of 2 film spaces normal to axis of tube = 1.488 square inches for 7/64-inch film and 2.290 square inches for 3/16-inch film.

D = Density of river water in pounds per cubic foot corresponding to items 7 and 10.

30. $\frac{\text{Item 12}}{\text{A}}$,

where A is the area in square feet of water-heating surface = 21.33 square feet for 7/64-inch film and 20.45 square feet for the 3/16-inch film.

31. $t = t_s - \frac{t_1 + t_o}{2}$

t = Average arithmetical temperature difference,
t_s = Temperature of saturated steam = item 20,
t_1 = Temperature of inlet water = item 7,
t_o = Temperature of outlet water = item 8.

32. $t_m = \frac{t_o - t_1}{\text{Log}_e \left(\frac{t_s - t_1}{t_s - t_o}\right)}$.

t_m = Mean geometrical temperature difference,
t_s, t_o and t_1 same as in item 30.

33. $\frac{\text{Item 30}}{\text{Item 31}}$.

34. $\frac{\text{Item 30}}{\text{Item 32}}$.

35. $\frac{\text{Item 28}}{\text{Item 6}} \times 1{,}000$.

36. $\frac{\text{Item 6}}{\text{A}}$,

where A is the water heating surface in square feet as given in item 30 above.

37. $\frac{\text{Item 12}}{\text{Item 26}}$.

38. $\frac{\text{Item 6}}{2}$.

STEAM TURBINE BLADE FASTENINGS.

By Jas. A. Capstaff, New York Shipbuilding Company, Camden, N. J.

So far as we are aware the era of the steam turbine dates back to 120 B. C., when Hero, of Alexandria, constructed his reaction wheel. Little is known of the use to which his turbine was put, but we can be pretty safe in saying that it amounted to little more than a philosopher's toy.

About the year 1630 A. D. Branca, an Italian, conceived the idea of utilizing the kinetic energy of steam and built a pure impulse turbine with a view to supplying power to take the place of that of man and beast.

The nucleus of the great prime mover of today lay dormant until the 19th century, when the possibilities of a velocity engine again revealed itself and engineers were beginning to appreciate the value of a rotary steam motor.

Wm. Gilman and others tackled the subject, but their lack of knowledge of the principles of steam in motion left their turbines little further than in their experimental stages.

Sir Charles A. Parsons' name stands out before us as the real pioneer of the steam turbine. His first patents were filed in the year 1884. Claims on blade fastenings have occupied no mean place in the patent records since that time, and it is with the most important of these I propose to deal in this paper.

All along the line turbine engineers have been striving to increase the efficiency of their engines. Repeated tests have proven that the efficiency varies with the blade speed and the number of rows of blades, and that the number of rows of blades varies inversely as the square of the blade speed for a

given efficiency. That is to say, if a turbine running at 600 revolutions per minute requires 64 rows of blades to show a certain efficiency, another turbine having the same mean blade diameter and the same height of blades, would only require 8 rows of blades, if running at 1,200 revolutions per minute for the same efficiency. From this it will be seen that increasing the number of revolutions per minute offers increased efficiency with smaller turbines, and the tendency is to develop along these lines.

The adoption of reduction gearing opens up a bright future for the further development of the steam turbine. With the direct drive the revolutions of the turbine have to be kept low for propeller considerations; but with gearing the turbine can be designed as an independent unit, without regard to the propeller revolutions. The question is how far can we go on increasing the peripheral speed of the blades and reducing the size of the turbine. For the answer we must look to the strength of the materials used in their manufacture. Theoretically the limit would be reached when the mean blade speed equals between .75 and .9, that of the steam in the case of a Parsons reaction turbine, and .5 in the case of a pure impulse turbine. Quite a common pressure drop through a turbine is from 265 pounds per square inch down to 28 inches vacuum. This drop in pressure represents a steam velocity of 4,200 feet per second.

The possibility of ever reaching the theoretical blade limit is indeed remote, but the tendency is to go on increasing the blade speed. It will be of interest to note here that Messrs. Parsons seldom exceed 550 feet per second at the mean diameter of the last expansion; Curtis as high as 700 feet per second, and DeLaval 1,375 feet per second; all a long way behind the theoretical most efficient speed. With the increase of the revolutions if the diameter is kept down the blade heights must run up. This cannot be done without taxing the strength of the blade and its fastenings.

In the turbines of today it is highly essential to bind a number of the blades in a row together, either by means of a shroud-

ing or lacing wires, in order to strengthen them against the fatiguing effect of the high period of vibration set up in the blade when running.

It would, of course, be desirable to have this shrouding or lacing, as the case may be, continuous, but, owing to expansion troubles due to the use of different metals for the turbine rotor and the blades, this is impracticable. Provision must be made to take care of the expansion. Previous to the adoption of the blade ties, blading gave no end of trouble through breaking at the roots. In passing I might mention that shrouding is used on impulse turbines to serve a double purpose. Besides tying the blades it prevents the lateral spreading of the steam jet on its passage across the turbine blade. Shrouding is not absolutely essential on reaction turbines, as the radial clearances between the rotor-blade tips and the cylinder walls are very fine; but one form of binding must be used. Channel shrouding is used by some builders as a safeguard against damage through-blade stripping. At the time of its conception blade stripping was of common occurrence; if one single rotor blade fouled with the cylinder wall the destruction of at least the rows in which it was contained was almost certain.

For some years now Parsons have adopted the clever idea of thin tipping their turbine blades for a small portion of their length, and should a revolving blade creep far enough from its root fastening to foul the casing the sharpened tip of the blade will simply be turned over and the rest of the row will not be affected.

In recent years impulse turbines have come to the front. The writer believes that they have much to recommend them, and when the mechanical design receives the attention it deserves, more will be heard of them in naval work. The fact that Messrs. Parsons have incorporated an impulse element in their reaction turbine lends some color to this remark.

With the development of the impulse turbine in particular the tendency to speed up is most marked, and this is bringing into being blades of much heavier section than have hitherto been used. Heavier blades and higher velocities keep the de-

signer's mind constantly working in the endeavor to provide a root fastening strong enough and reliable enough to take care of the increasing loads imposed upon it. That this is no easy proposition must be admitted.

Perhaps no field in the design of the steam turbine has been so much exploited as that of blade fastenings, as a survey of the patent records will show. Many ingenious methods have been devised from time to time, the most important of which will be detailed later. Some of the fastenings used today, though highly suitable to meet the conditions existing at the time they were devised, must soon be consigned to the scrap heap or museum, and give place to a fastening of unquestionably superior strength. Too much stress cannot be laid upon the importance of strength. No blade fastening can be too strong. Most builders have, at some time or other, made up a test piece of their particular form of blade fastening and had it pulled to destruction in a testing machine, and demonstrated that it required a pull far exceeding the centrifugal force to which it would be subject when running to destroy it. But, while a carefully-prepared test piece pulled in a testing machine affords excellent material for comparison with other methods, it is poor practice to use the test results as a criterion for running conditions. The unknown quantity enters too largely into this equation. To repeat myself, "The direct pulling strength of a blade fastening ought to be as great as possible, but not only that, the blade must be rigidly held in every direction with no possible clearance to allow of movement."

Perfect rigidity is the hall-mark of excellence in a blade fastening, especially so in an axial direction, for upon this, to a very large extent, depends the life and reliability of the fastening. If there should be any initial clearance between the blades and the sides of the grooves the vibration to which the blades are subject when running under steam may be the means of working the blades loose. Machine fits cannot be relied upon to ensure the desired side fit, and caulking must be resorted to. Without caulking we have no guarantee that the

blades completely fill the width of the groove. Caulking is the best known method of assuring biting contact; but while the writer is a keen advocate of its use to ensure rigidity he is not in favor of a blade fastening which relies upon caulking for its direct pulling strength. Caulking should be employed only as a means of eliminating freedom of movement. Heavy caulking is to be avoided on turbine rotors and discs. It distorts them and calls for increased pitches of rows and heavier rotors and discs. In high-speed turbines it is desirable, for reasons of efficiency, to keep down the axial pitch of the rows as much as possible. To do so with a blade fastening of questionable strength and rigidity would be courting disaster.

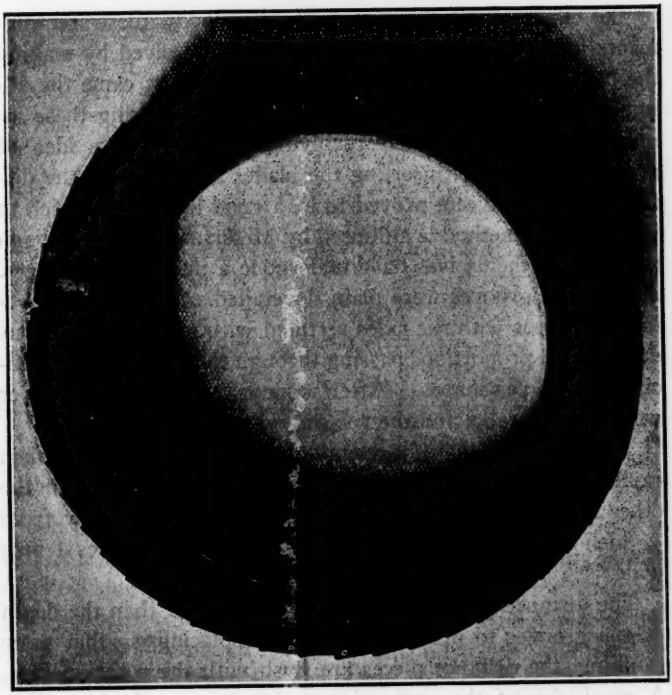


FIG. 1.

Before proceeding with a general description of the various methods of blade fastenings, mention should be made of the desirability of giving liberal depth to the groove, as the blade has to rely upon the depth of its fastening for its steadyment.

The blades of the earliest machines built by Messrs. Parsons were made in complete rings which were keyed onto the turbine shaft. The blades were formed by saw cutting the outer circumference of the rings and twisting the edges with pinchers to the desired angles. Experience went to show that while this method might provide the correct angles of entrance and discharge, it left much to be desired in the way of the gradual changes of the course of the steam on its passage across the blade and the smooth polished surface so essential to the efficiency of the turbine. To embody these requirements in a blade was a problem which was solved by making the blades in lengths and sawing to size. Next came the all-important question as to the best means of securing these innumerable blades to their revolving elements. The idea of grooving the rotor, inserting the blades and caulking the distance pieces between, proved to be a good line along which to work. This method is still used by Messrs. Parsons, although the grooves themselves have undergone a number of changes. At first the grooves were plain dovetailed. Then came dovetailed grooves with the sides serrated with grooves of various designs and proportions passing through the rectangular, semicircular and vee shapes. After some years the grooves proper were made parallel and the vee-shaped serrations adopted.

Tests showed that the metal of the distance pieces when caulked flowed more freely into the vee serrations than the other types mentioned. The depth of the grooves vary from ¼ inch for blades ¼ inch wide, to ⅝ inch for blades 1 inch wide.

The distance pieces are cut 1/32 inch longer than the depth of the grooves to provide metal for displacing, so that after caulking, the distance pieces are flush with the rotor surface. A good idea of the proportion a Parsons standard reaction blade fastening for rotors may be seen in Fig. 2.

Fig. 3 shows the Parsons' method as applied to impulse blades. Owing to the heavier sections the rotor grooves are made deeper than those for the same width of reaction blades, and in order to ensure contact at the lower part of the root the distance pieces are double caulked, or, in other words, inserted in two pieces—the lower being caulked hard up before the top piece is inserted. The fronts and backs of the blades at the roots are serrated with vee-shaped serrations into

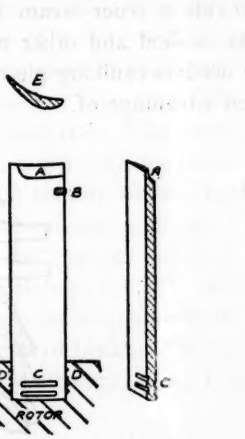


FIG. 2
PARSONS STANDARD
REACTION BLADE
'A' THIN TIPPING. 'B' LACING STRIP
'C' INDENTATIONS 'D' SERRATIONS IN ROTOR
'E' BLADE SECTION.

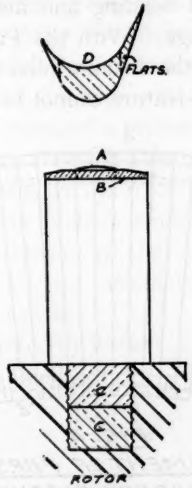


FIG. 3
PARSONS STANDARD
IMPULSE BLADE.
'A' RIVETING 'B' SHROUDING.
'C' DOUBLE CAULKED DISTANCE PIECES
BETWEEN BLADES. 'D' BLADE SECTION.

which metal of the distance piece is driven when caulked. It will be noticed in this type of fastening that the holding power of the blade is solely dependent upon its adjacent distance piece; whereas in the Parsons reaction fastening, where the blades are singly inserted, the blades contribute to the holding power, due to the fact that the rotor groove is made narrower than the width of the blades, and the blades are fullered through a degree or two in driving them together in the

groove, thereby making a biting contact between the edges of the root of the blades and the side of the rotor groove; the side flats existing at the roots after cutting back the fine edges of the blades do not permit of this being done in the case of impulse blades.

It is the practice with a good many impulse-turbine builders to allow the distance piece to project up the blade beyond the top of the rotor groove to give the blade increased strength against bending and also provide a truer steam line across the stage. With the Parsons method and other methods in which the distance pieces are used as caulking pieces, this desirable feature cannot be taken advantage of.

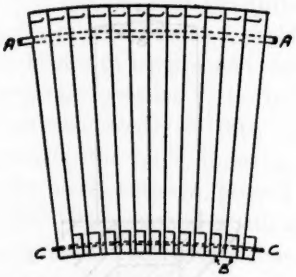


FIG. 4
SEGMENT OF PARSONS ROSARY BLADING
"A" LACING STRIP. "B" DISTANCE PIECES. "C" WIRE THROUGH ROOT.

FIG. 5
PARSONS COCKSCOMB ROOT.

The Parsons method has proved itself to be an excellent fastening for reaction blades running up to 550 feet per second, and suitable for impulse turbines of low blade speed, but it has yet to prove itself capable of taking care of the higher blade speeds suggested as a means of developing maximum efficiency with minimum dimensions. You are referred to the table further on in this paper for test fixtures on the Parsons blade fastenings. In order to shorten the time of erection and facilitate repairs, Messrs. Parsons build up their blading in segments such as shown in Fig. 4.

The blades and distance pieces have a hole drilled through them near the base and are assembled in a former with a wire threaded through them, which is turned over at the ends to hold the segment together until ready to put into rotor. This type goes by the name of the Rosary Blading. An earlier attempt at building up this blading in segments is shown in Fig. 5 and was known as the cock's comb method. It derived its name from the shape of its foundation ring, which was milled out at intervals to receive the blades, giving to it a close resemblance of a long cock's comb. After the blades were inserted the points of the comb were knocked over and thus held the blades in position. The life of this type was a comparatively short one. The fact that it required a groove about 25 per cent. wider than the blade was an objectionable feature.

Segmental blading after being inserted in the rotor groove must be individually caulked. This individual caulking required of the Parsons method, constitutes, to the writer's mind, a weakness, for the reliability of the fastening is dependent upon the scrupulosity of the blader.

Other types of blade fastenings patented by Messrs. Parsons and his associates are shown in Figs. 6 and 7.:

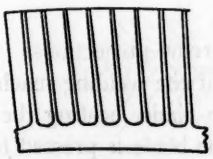


FIG. 6
PART SEGMENT OF EARLY PARSONS BLADING CUT FROM SOLID PIECE OF DELTA METAL.

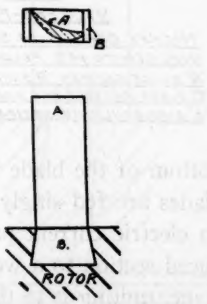


FIG. 7
PARSONS BLADE AND DISTANCE PIECE COMBINED.
'A' BLADE SECTION 'B' BASE.

In Fig. 7, the blades and distance piece are in one and are let into a dovetailed groove, and the distance-piece projection is caulked in the usual Parsons method.

An ingenious blade fastening is that of Ferranti. In its simplest form the blades are made of nickel-coated mild steel and electrically welded to the turbine rotor. For marine work the blades are welded to mild-steel foundation rings, which are secured to the rotors; the object being to enable spares to be carried and fitted easily. Such an arrangement is shown in Fig. 8. The rotor or foundation ring is serrated and the

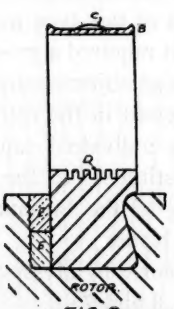


FIG. 8
VICKERS-FERRANTI
NICKEL COATED M.S. BLADE ELECTRICALLY WELDED TO M.S. FOUNDATION RING.
'A' BLADE SECTION. 'B' SHROUDING. 'C' RIVETING.
'D' BLADE PROJECTIONS AND BASE SERRATIONS.
'E' SIDE CAULKING STRIPS.

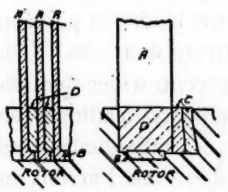


FIG. 9.
WESTINGHOUSE BLADE.
'A' BLADE. 'B' BLADE UPSET
'C' SIDE LOCKING RINGS.
'D' DISTANCE PIECE BETWEEN BLADES.

bottom of the blade is cut to form narrow projections. The blades are fed singly into the holder of the welding machine, an electric current passed through the blade, heating the reduced section to a welding heat and the blade is pressed hard home, uniting it to the rotor or foundation ring. Some time ago the writer had the pleasure of making some pulling tests on this type of fastening and found the weld to be as strong as the blade itself. If reblading had not to be taken into consideration and the blades were welded directly to the rotor, we would have a fastening equal in strength to the blade. As mentioned above, for marine work it is highly desirable to

weld the blades on to segments and secure these segments to the rotor. This constitutes the weakest link in the chain. In the list of tests further on in this paper 100 per cent. efficiency was assumed at the weld, and a piece of mild steel of a thickness equal to one blade and a distance was secured to the holder by means of a brass side caulking piece, and pulled to determine the holding power.

The Westinghouse Machine Company, of Pittsburgh, are the patentees of the fastening shown in Fig. 9.

As in the Parsons method, the distance piece forms the anchor. The upset at the bottom of the blade provides a reliable connection between the blade and the distance pieces. The side pieces provide side grip for the blades and secure the distance pieces. They are made up of tapered wedges which facilitate assemblage and repairs.

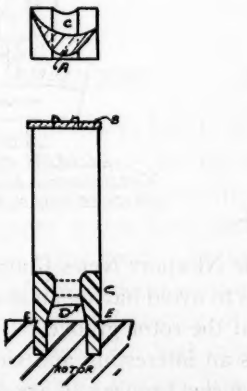


FIG. 10

FORE RIVER FASTENING

"A" SECTION THROUGH BLADE. "B" SHROUDING. "C" CHANNEL BASE. "D" RIVETING. "E" ROTOR CAULKING.

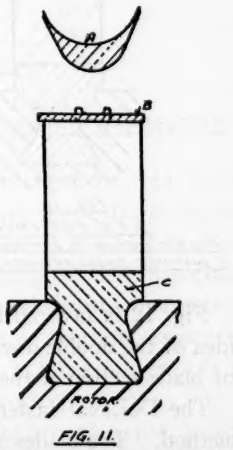


FIG. 11.

PLAIN DOVE-TAIL FASTENING

"A" BLADE SECTION. "B" SHROUDING. "C" DISTANCE PIECES BETWEEN BLADES.

Fig. 10 shows the Fore River arrangement. It consists of lengths of steel channel-punched at intervals to receive the blades, which are riveted in. The segments of blading are then bent to the curvature of the rotor and inserted in grooves

in the rotor. Caulking of the sides of the grooves holds the blading in position. It is a particularly clever scheme, but in its present form the part of the channel which receives the blades is not thick enough, in the writer's opinion, to provide the length of steadyment necessary to keep down the blade deflection within safe limits. With such a short steadyment the demands made upon the riveting are severe.

The plain dovetail fastening shown in Fig. 11 is now being used by the Fore River Company.

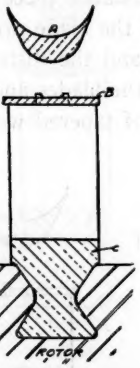


FIG. 12
"A. E. G." FASTENING.
A BLADE SECTION. B. SHROUDING.
C DISTANCE PIECES BETWEEN BLADES

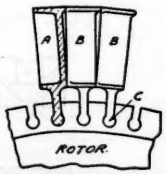


FIG. 13
DE-LAVAL FASTENING.
"A" SECTION THROUGH BLADE.
"B" OUTSIDE VIEW OF BLADE. "C" BULB.

Fig. 12 is favored by The Newport News Company. The idea of the double dovetail is to avoid having an abrupt change of blade section at the top of the rotor groove.

The DeLaval fastening is an interesting and well tried-out method. The blades are of nickel bronze and are drop forged with shrouding and roots in one piece. The roots are bulb shaped and are slightly tapered transversely. The rim of the rotor disc is slotted at intervals to receive the blades, which are driven in as shown in Fig. 13.

The blades are not riveted to the shrouding nor are they riveted to the disc. The driving fit used for holding the bulb shanks makes the replacement of blades a simple matter, with-

out danger of distorting the wheel or throwing it out of balance. This result depends upon careful and exact fitting.

The fastening shown in Fig. 15, has not, so far as I know, been adopted on any commercial unit, but its ingenuity is worthy of mention. A dovetailed groove is cut in the rotor with its base forming a wedge and the pressing home of the blades and distance pieces automatically spreads them into the dovetail, filling up the groove. Dr. Jude, of Manchester, England, claims the originality of this scheme.

FIG. 14
EARLY DE-LAVAL FASTENING

FIG. 15
FASTENING WITH WEDGE BASE

Much credit is due to Mr. L. D. Lovekin, of the New York Shipbuilding Company, for the excellent blade fastening he has recently devised, especially in view of the fact that so many designers' minds have been working on this important detail for so many years. Its sound mechanical construction is worthy of the serious consideration of turbine engineers looking for a rigid fastening of great holding power. It is particularly suitable for high-impulse work.

In Mr. Lovekin's method, Fig. 16, the rotor or disc, as the case may be, is grooved with a plain dovetailed groove into which is inserted a tongued locking piece about twelve inches long. Alternately the blades and the distance pieces, whose roots are milled to a dovetail on one side and a recess on the other, are threaded on to this locking piece until the length is made up. Another piece of locking strip is inserted and more blades and distance pieces threaded on and the operation re-

peated until the entire circumference of the groove is filled up with the exception of a piece of locking strip equal in length to two blades and two distance pieces. This length is closed up as shown in Fig. 17 or 18. After assembling one complete row the locking ring is caulked well down to make sure that the blading completely fills the groove. Mr. Lovekin's side-locking piece is so designed that in the process of caulking the roots of the blades and the distance pieces are first brought in contact with the bottom of the groove and then the blades and distance pieces are forced bodily against the side of the rotor groove. It will be noticed that the bottom of the locking piece is undercut, so that, in caulking, a spreading

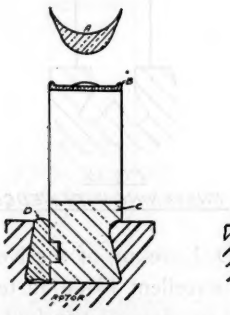


FIG. 16.
LOVEKIN FASTENING
'A' BLADE SECTION. 'B' SHROUDING.
'C' DISTANCE PIECES BETWEEN BLADES & LOCKING PIECE.

FIG. 17.

FIG. 18.
CLOSING-UP PIECES FOR LOVEKIN METHOD.

action takes place at the bottom and tightens the lower portion of the blading as well as the upper. The idea has proved very effective. The first three stages of the U. S. torpedo-boat destroyer *Downes'* turbines were bladed in accordance with the Lovekin method, and the finished result presented rows of blading as perfectly in line as could have been obtained had each blade been trued with a square.

The admirable features of the Lovekin method are:

1. Its rigidity;
2. Its great direct pulling strength;

3. The fact that heavy caulking has not to be resorted to to give it its direct pulling strength;

4. That full advantage has been taken of the best known method of assuring rigidity, namely, side caulking;

5. The blading operation is quick, and so simple that the services of experienced bladers are not essential;

6. It is fool-proof;

7. No special recesses are required around the circumference of the rotor grooves for the insertion of the blades—an item of importance in reblading;

8. Extended distance pieces, so valuable in strengthening the blades against bending and providing a good steam line across the stage, can be used;

9. Perfect alignment of the blades in a row is assured;

10. There is no abrupt change of section of blade at the surface of the rotor.

Realizing the many advantages of the Ferranti method, such as eliminating variable circumferential expansion and obtaining an absolute fit of all sectional-base rings by side caulking, "instead of depending on a machine fit which is impossible in manufacturing," caused Mr. Lovekin to consider the Ferranti method as the most advanced or ideal method yet produced. Therefore, he set to work to design a means of holding turbine blades that would have all of the advantages of the Ferranti method without the disadvantage of welding blades to the base rings, which, as is well known, is difficult and costly and beyond the scope of the engineer at sea to repair in case of breakage. The Lovekin method "when made of Monel metal blades with distance pieces combined and having a Monel metal locking ring, which is caulked solidly after all blades have been fitted," enables a result to be obtained that has heretofore been unheard of. The coefficient of expansion is almost the same as that of steel. Then, again, the simplicity of manufacture is such that anyone can blade or reblade a turbine, whether at sea or in the machine shops. This is of paramount importance.

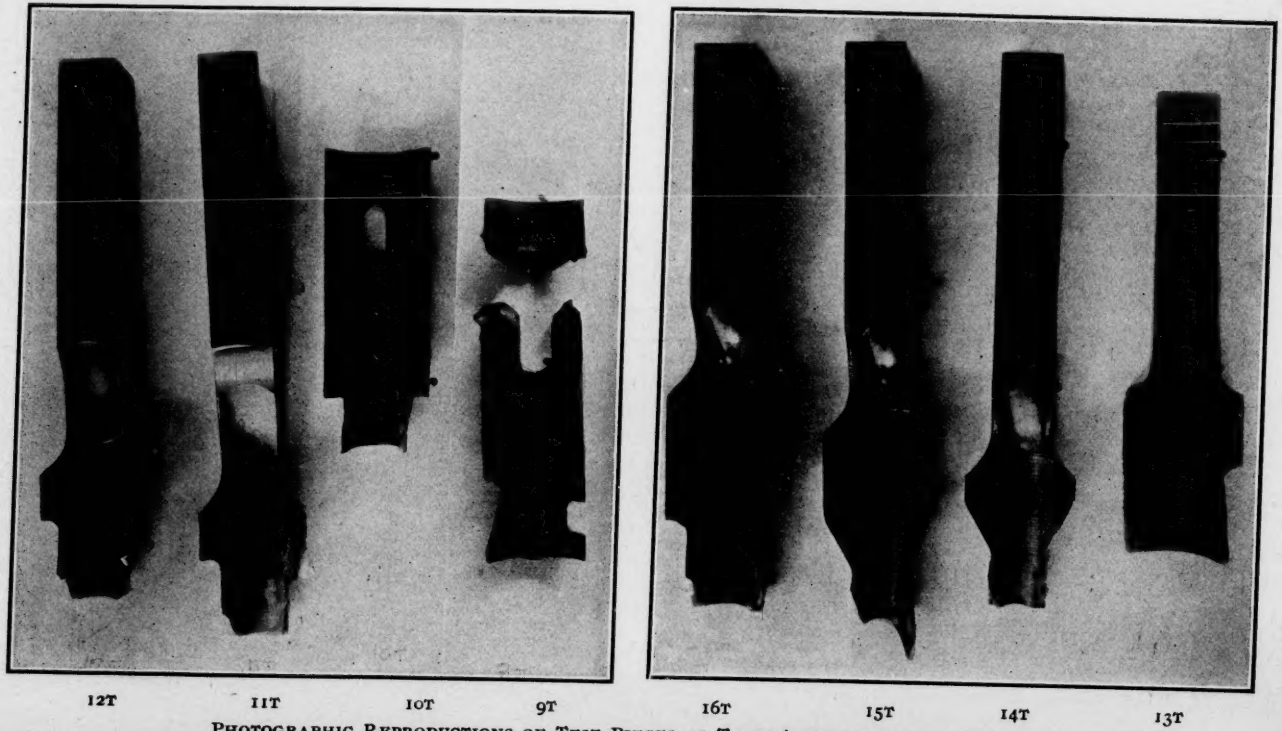
Realizing that it is unnecessary in the lower circumferential speeds such as are used on marine turbines of the direct-drive type (or without the interposition of gearing) to provide the maximum strength of blading, tests were made with the Lovekin method of blading with various modifications. While the same sectional view as shown in Fig. 16 held good for all the test pieces, the actual construction differed. Test No. 7 was made with a brass blade and distance piece combined, using a locking piece of brass. Test No. 12 was made with a brass blade and separate brass distance pieces; but instead of milling out the blade recess the metal was pressed over to engage in its adjacent distance piece. Two other test pieces were made up having a brass blade and separate brass distance pieces, with a ¼-inch diameter wire through the roots, as shown in Nos. 9 and 11. Tests No. 1-A and No. 2 were made with a brass blade and separate brass distance pieces. There are also shown test pieces of the Parsons double-caulked method, No. 8; the Fore River method, No. 6; the fastening as used on the 1st stage of the Argentine battleship *Moreno's* turbines, No. 10; the AEG, Nos. 14 and 15, and the plain dovetail, No. 16. The disadvantage of these three latter methods lies in the great accuracy required in manufacture; while the disadvantage of the Parsons method is its limited application for high centrifugal speeds.

The blade sections used were the latest Fore River Curtis 1.3 inches wide, with the exception of Test No. 6, which was 1 inch wide. It will be noticed that the majority of the test pieces were cut from the bar, the object being to provide a suitable end to accommodate the testing-machine grips.

Figs. 1A.T. to 16 T, are photographic reproductions of the test pieces as they appeared after pulling.

In all the Lovekin tests the metal below the recess and the metal at the root of the dovetail was drawn and eventually torn away. The bottoms and sides of the tongues on the locking pieces were locally indented about .01 inch in way of the blades.

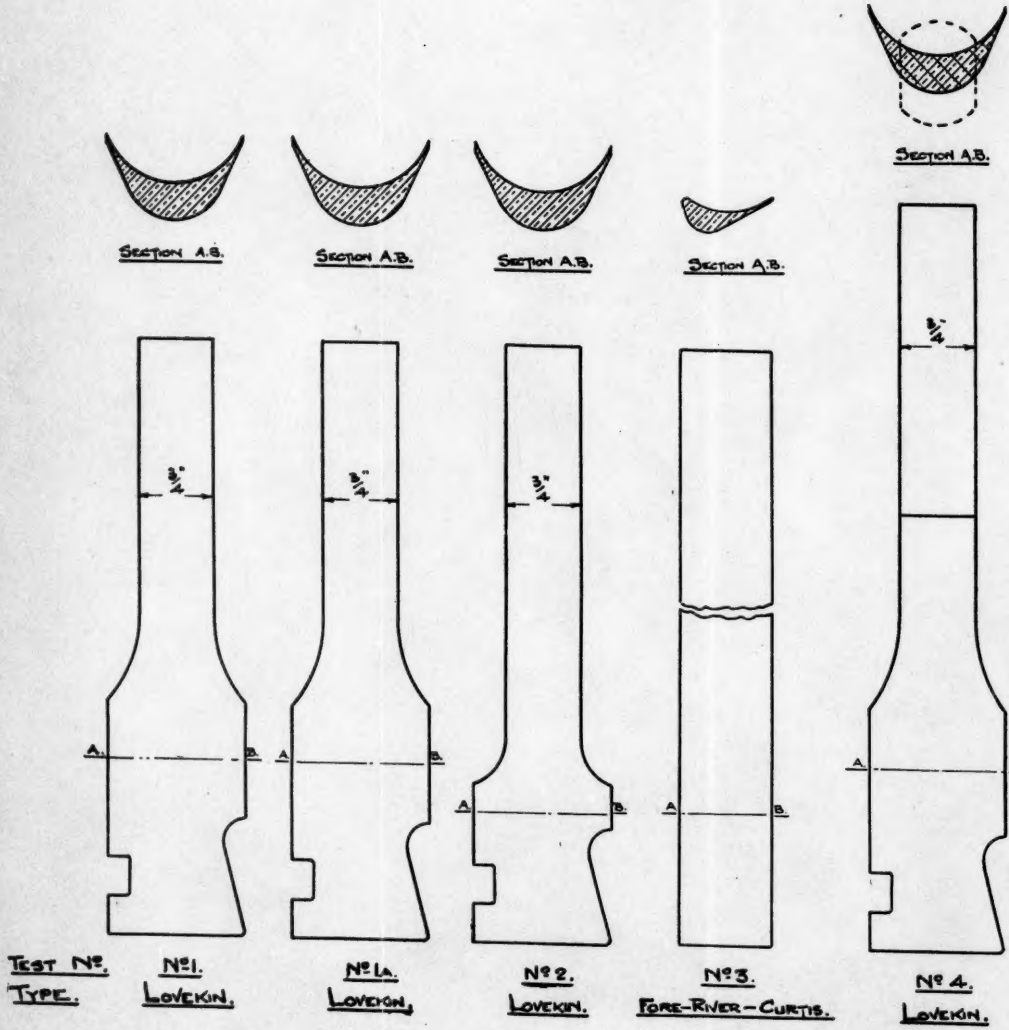
PHOTOGRAPHIC REPRODUCTIONS OF TEST PIECES AS THEY APPEARED AFTER PULLING.

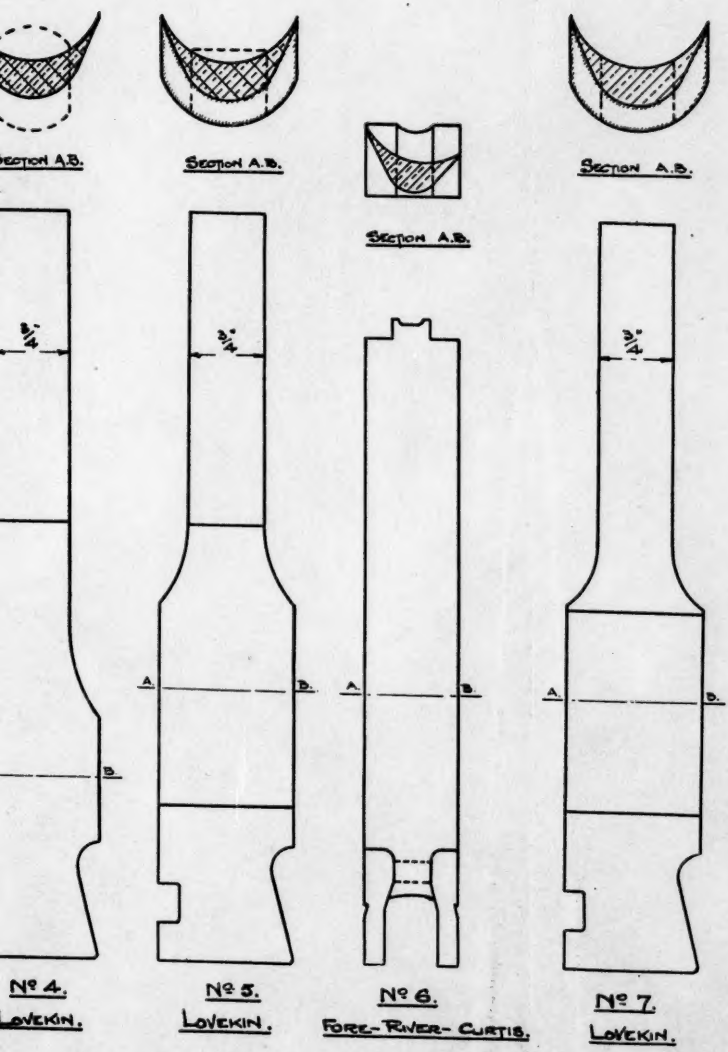


12T 11T 10T 9T 16T 15T 14T 13T

PHOTOGRAPHIC REPRODUCTIONS OF TEST PIECES AS THEY APPEARED AFTER PULLING.

PHOTOGRAPHIC REPRODUCTIONS OF TEST PIECES AS THEY APPEARED AFTER PULLING.


SECTION A.B.
SECTION A.B.
SECTION A.B.
SECTION A.B.
SECTION A.B.
3/4"
3/4"
3/4"
3/4"
A.
B.
TEST Nº.
TYPE.
Nº1.
LOVEKIN.
Nº1A.
LOVEKIN.
Nº2.
LOVEKIN.
Nº3.
FORE-RIVER-CURTIS.
Nº4.
LOVEKIN.



SEE VIEW ABOVE. FIG. 20.

DETAIL OF TEST PIECES BEFORE PULLING.

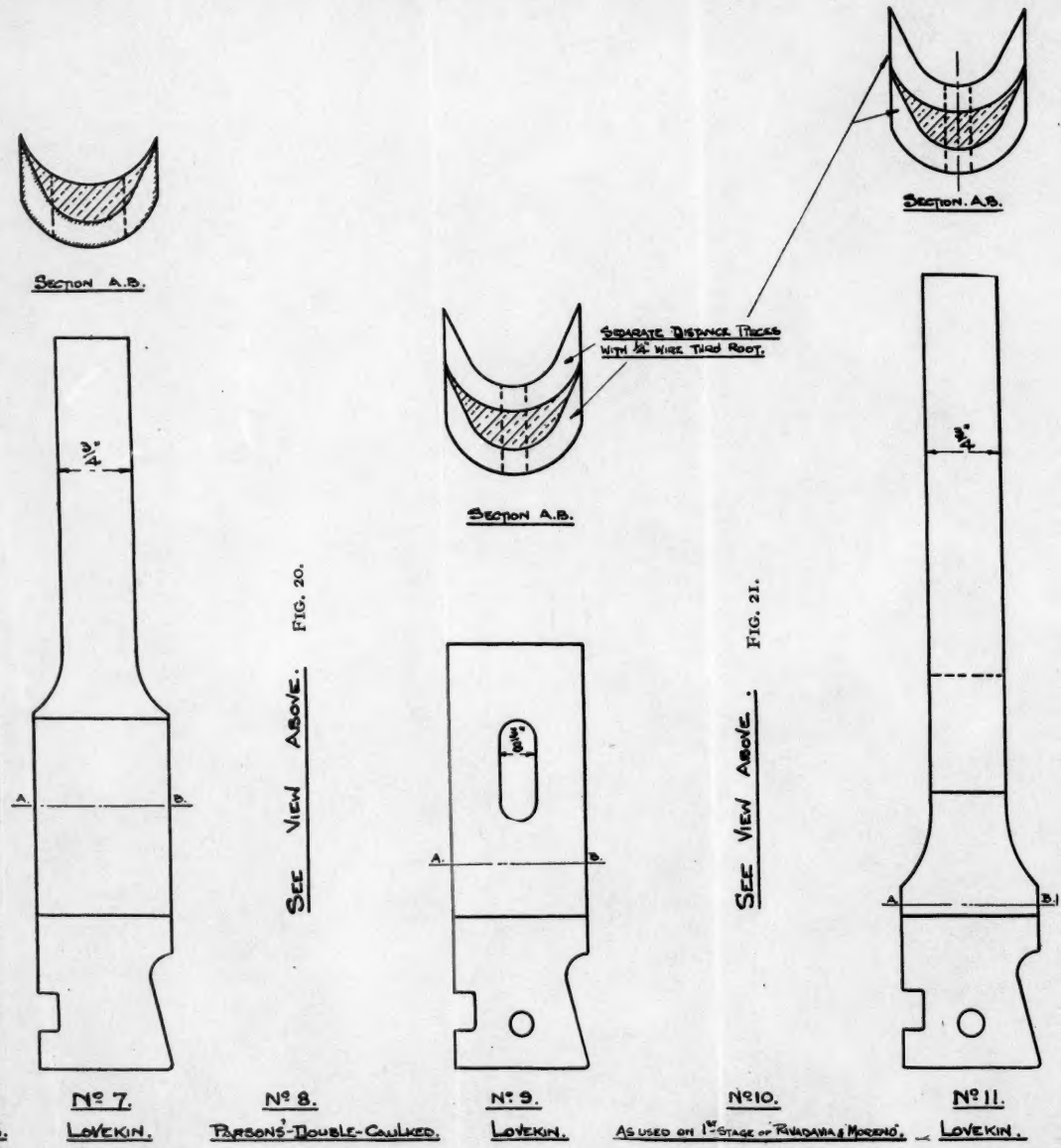
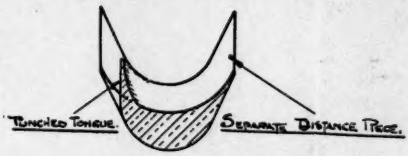


FIG. 20.

FIG. 21.

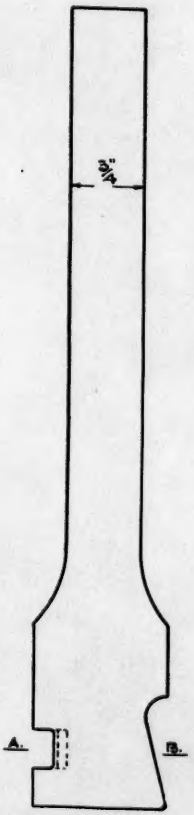
DETAIL OF TEST PIECES BEFORE PULLING.

N. A.B.



Section A.B.

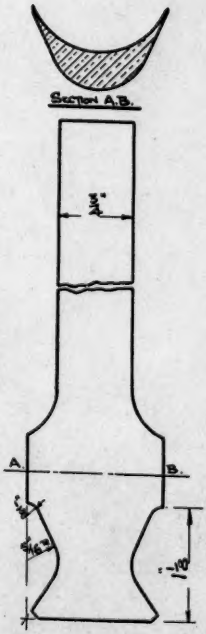
Nº 11.
VEKIN.



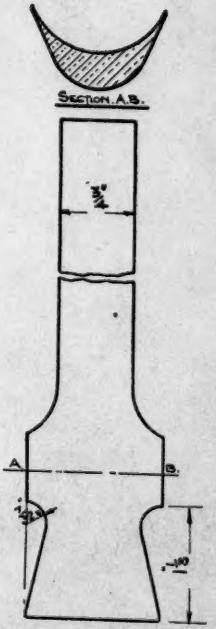
Nº 12.
Lovekin - Capstaff.

See View Above. Fig. 22.

Nº 13.
Vickers - Ferranti.



14 & 15
A.E.G.



16.
Plain Dovetail.

RESULTS OF TESTS ON VARIOUS METHODS.

Test No.	Piece No.	Description.	Method.	Pounds pull required.		Locking piece.
				To start blade moving.	To pull blade entirely out.	
13	13T	Steel foundation piece with side caulking pieces.....	Vickers-Ferranti.	.....	10,700	Brass.
5	5T	Monel blade, solid, with distance piece.....	Lovekin.....	16,900	25,200	Monel.
7	7T	Brass blade, solid, with distance piece.....	Lovekin.....	14,000	17,500	Brass.
4	4T	Monel blade with separate Monel distance pieces.....	Lovekin.....	9,700	12,300	Monel.
11	11T	Brass blade with separate D.P.'s with ¼-inch diameter wire through root.	Lovekin.....	8,000	10,500	Brass.
9	9T	Brass blade with separate D.P.'s with ¼-inch diameter wire through root.	Lovekin.....	Broke at reduced	section with 9,800	Brass.
12	12T	Brass blade with punched tongue, with separate brass distance pieces.	Lovekin.....	8,500	10,100	Brass.
2	2T	Brass blade with separate brass distance pieces.....	Lovekin.....	5,000	7,000	Brass.
1A	1AT	Brass blade with separate brass distance pieces.....	Lovekin.....	5,000	6,800	Brass.
6	6T	1-inch brass blade riveted to M. S. channel base.....	Fore River.	1,500	2,400	...
8	8T	Brass blade with double caulked distance pieces.....	Parsons.....	3,400	A continued diminishing load.	...
10	10T	Brass blade with separate distance pieces.....	*Moreno*.....	3,700	4,700	...
14	14T	Brass blade with separate distance pieces.....	A. E. G.....	1,500	9,400	...
15	15T	Brass blade with separate distance pieces.....	A. E. G.....	2,500	8,700	...
16	16T	Brass blade with separate distance pieces.....	Plain dovetail.	2,400	8,700	...

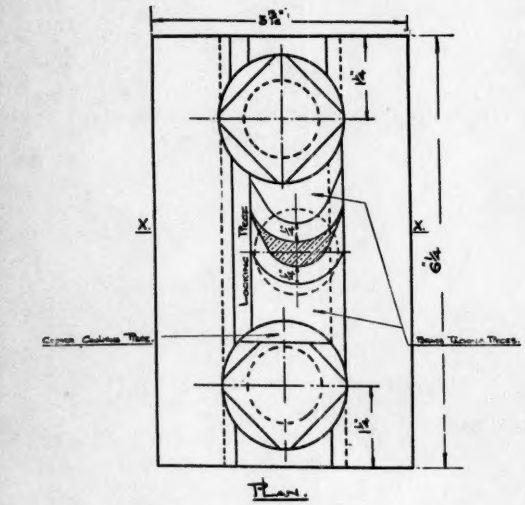
All these test pieces were made up at the New York Shipbuilding Co. and pulled to destruction at Wm. Sellers & Co.'s works on their 100,000-pound Emery testing machine, in the presence of Mr. W. R. Church, of Wm. Sellers, and Messrs. F. F. Kauffman and Jas. A. Capstaff, of New York Shibuilding Company.

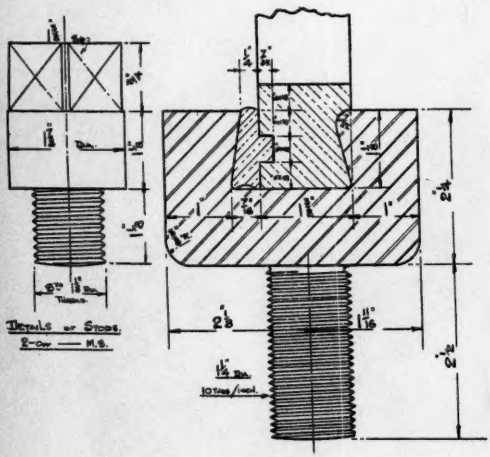
In the Parsons' test the metal of the distance pieces, which was caulked into the vee serrations of the blades and holder grooves, was sheered off.

In the Vickers-Ferranti test the root of the dovetail was drawn, and sheering of the caulked metal took place as in the Parsons test.

The riveted root of the Fore River fastening was drawn parallel.

The other specimens pulled out after drawing and sheering, as will be clearly seen in the photograph. The test-piece holder was of mild steel ,in accordance with Figs. 19 to 22. The projecting piece of the holder was screwed into the weighing head and the blade shank gripped in the jaws of the straining head of the testing machine. The load was applied by the hydraulic straining cylinder, which received its power from a suitable fluid-pressure supply. The essential peculiarity of the "Emery" testing machine is the method by which the stress produced upon the piece being tested is conveyed to the scale and accurately weighed by mechanism that is entirely frictionless, and hence responds equally to the same increment of load regardless of the amount of strain upon the specimen. This result is accomplished by receiving the load upon a flexible diaphragm in a closed cylinder on a hydraulic support. One of these is placed in the scale and a proportionately large one in the machine itself. The pressure received on that in the machine forces liquid to the scale, through a tube, which, acting upon the diaphragm of much smaller area, produces sufficient motion to move the beam of the scale. The general scheme is indicated in Figs. 23 and 24, which shows merely the relation of parts, no attention being paid to proportions. In Fig. 24 "A" is the hydraulic support in the weighing head of the machine, which is connected through a metallic tube to the reducing chamber "B" in the scale. The pressure of the fluid upon the diaphragm in the reducing chamber is transmitted to the large lever "C," and thence through levers and connections to the indicator "F." Suspended from the lever "E" at suitable intervals are poise frames "N," which work

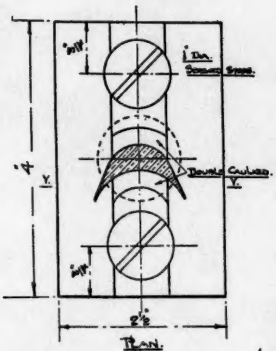


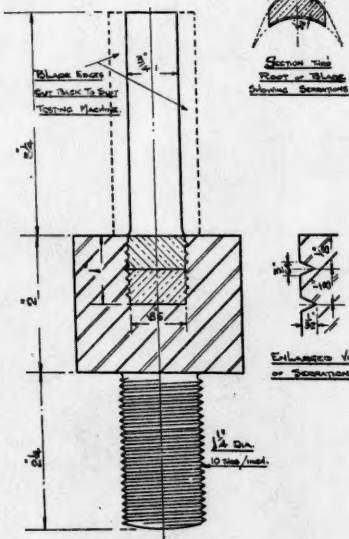


Details of Blade Holder.

As Used for Tests Nos 1, 1a, 2, 4, 5, 7, 9, 11, 12, Leveson.

Fig. 19.

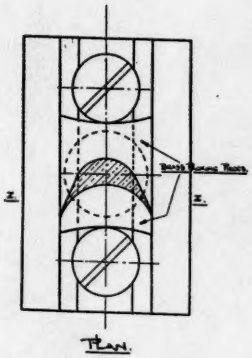




Details of B

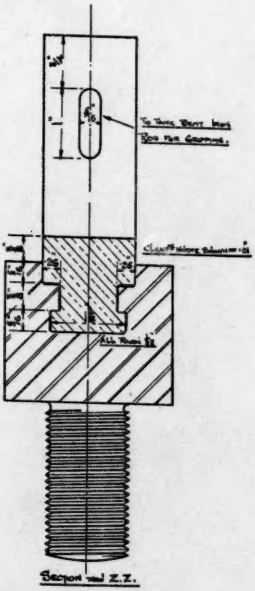
As Used for Test No 8.
Parsons Double Caulked.

Fig. 20.



PLAN.

Section thro' Root of Blade showing Serrations.



Enlarged View of Serrations.

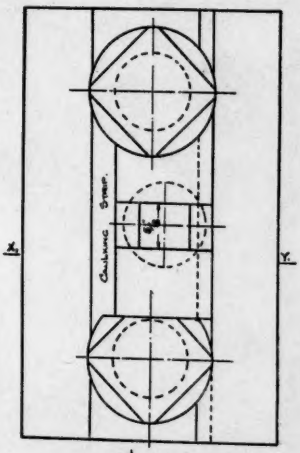
Section thro' Z.Z.

DETAILS OF BLADE HOLDER.

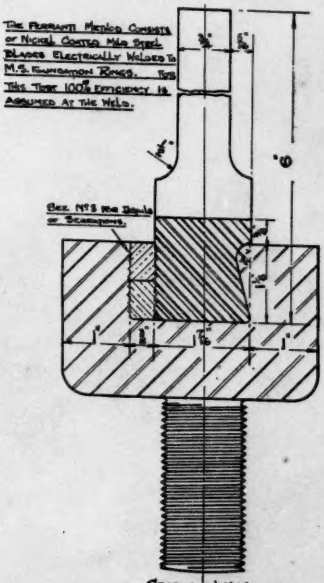
AS USED FOR TEST Nº 10.

"ROBSON'S & "MORENO" 1ST STAGE.

FIG. 21.



PLAN.


The Ferranti Method Consists of Nickel Coated Mild Steel Blades Electrically Welded to M.S. Flanged Rings. This Gives 100% Efficiency is Assumed at the Weld.
Recs. Nº 8 Mild Steel Blades or Segments.

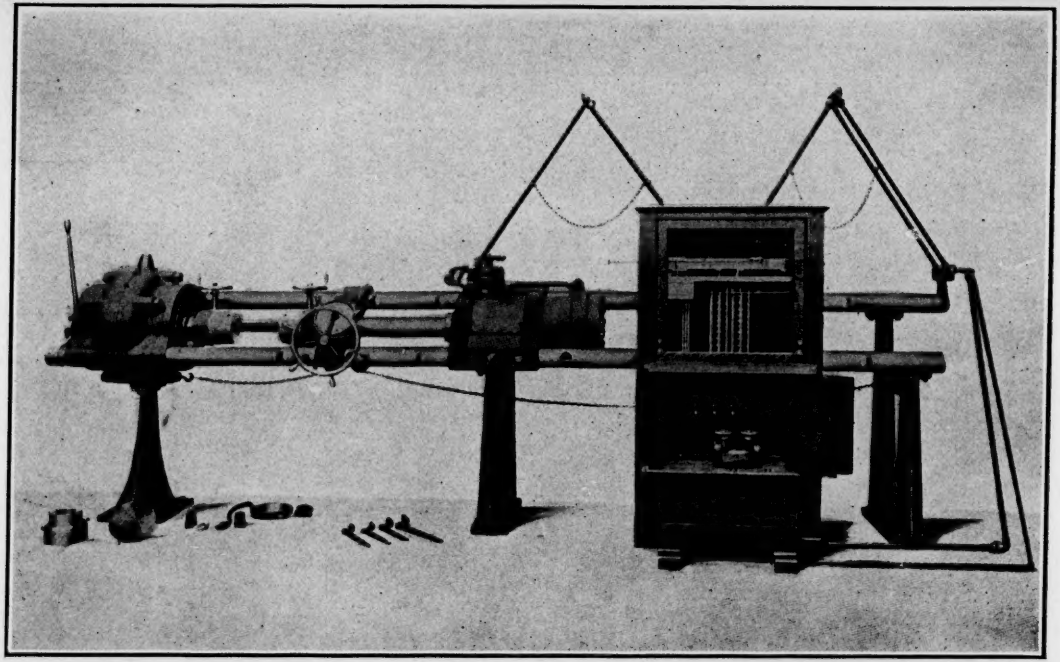
Section thro' X.Y.

DETAILS OF BLADE HOLDER.

AS USED FOR TEST Nº 13.

VICKERS – FERRANTI.

FIG. 22.



Capable of testing specimens 12 feet long in compression, and 8 feet, 4 inches long in tension. Maximum stroke of piston in straining head, 24 inches. Side bars arranged on incline for transverse testing, also facilitating access to the specimens. A hand wheel is provided for quickly moving the piston for entering and removing the specimens. Hydraulic scale shown in position. Tension holders and specimen in place. The compression platforms are shown lying upon the floor with other fixtures.

FIG. 23.—100,000-POUND TESTING MACHINE.

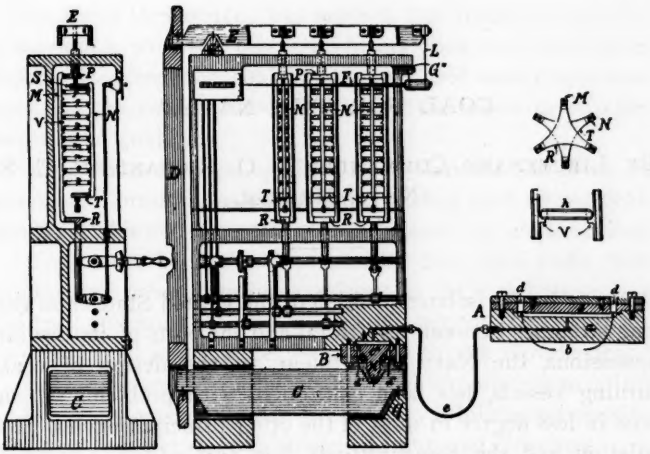


FIG. 24.—EMERY TESTING MACHINE WEIGHING MECHANISM.

in conjunction with the weight frames " M," in such a manner as to permit the weights to be applied in succession by the movement of the hand levers, until an exact balance is obtained. This load applied to the specimen may be read at any time from the numerals which appear in a convenient place on the scale. In place of knife edges, thin steel fulcrum plates are used, which deflect slightly with the vibration of the levers, without measurable friction. Each scale is calibrated with great accuracy, each weight being tested separately in position.

COAL FOR THE NAVY.*

By Lieutenant Commander J. O. Richardson, U. S. Navy, Member.

Owing to the isolated position of the United States and the long stretches between its littoral and the ports of its insular possessions, the Navy Department, in the design of coal-burning vessels, has been confronted with problems which exist in less degree in most of the other principal navies. Its isolation and the comparatively few and widely separated coaling stations made it necessary to devote especial attention to affording the ships the greatest possible coal-carrying capacity. To enable the necessary weight of coal to be carried, it has been necessary to limit to the utmost the weight of armor and of machinery, thereby sacrificing to some extent the vital qualities of protection and speed. The necessity of limiting the weight of, and space occupied by, the boilers has forced upon the Department the installation of boilers of light weight and compact design with limited combustion space.

In order to insure that the coal-burning vessels of the Navy will have the maximum steaming radius, it is necessary that they be supplied with coal that is suitable for use in the boilers on board these vessels and is the best coal obtainable for that purpose regardless of cost.

Section 3728 of the Revised Statutes of the United States provides that, "In purchasing fuel for the Navy, or for naval stations and yards, the Secretary of the Navy shall have power to discriminate and purchase, in such manner as he may deem proper, that kind of fuel which is best adapted to the purpose for which it is to be used."

* This article is largely compiled from the records of the Bureau of Steam Engineering and from bulletins issued by the Bureau of Mines.

The Navy Department has made it the practice to supply naval vessels with the best semi-bituminous coal obtainable. Strictly speaking, the coal is not purchased under specifications, but the coal delivered must meet the following requirements as to quality:

"Coal to be the best quality, Navy standard steaming, or equal, semi-bituminous, run of mine, with at least 40 per cent. lump, suitable and acceptable for the uses of the naval service.

"Coal must be dry and practically free from slate, dirt, sulphur and other impurities, subject to the usual inspection and test, and must weigh 2,240 pounds per ton, the weight to be determined in a manner satisfactory to the Government."

Navy standard steaming coal is coal supplied from mines on the Navy acceptable list, the delivery fulfilling the requirements as to moisture, percentage of lump and cleanness.

The Navy acceptable list is a list of mines that have supplied coal which has proved satisfactory for use on board naval vessels and whose output is controlled by reliable suppliers that may be depended upon to deliver the required amount of satisfactory coal under all conditions of the coal and labor market.

The question as to the reliability of the supplier is one of vital importance because it is necessary that the Navy be assured of an adequate supply of coal on short notice, and in actual practice it has been found that the large suppliers whose mines are on the acceptable list have supplied coal to the Navy to meet the full requirements when there was a large demand for coal at a price above the Navy contract price.

Mines are liable to suspension or elimination from this list if the deliveries of coal are the subject of complaint by the ships of the fleet or if the supplier fails to fulfill the conditions of his contract.

Each bidder on the naval coal contracts is required to supply the following information regarding the coal he proposes to furnish:

(*a*) Commercial name of coal.

(*b*) Name of mine or mines.

(*c*) Exact location of mine or mines.
(*d*) Name or other designation of the coal bed or beds.
(*e*) British thermal units per pound of "dry coal."
(*f*) Percentage of ash in "dry coal."
(*g*) Percentage of sulphur in "dry coal."
(*h*) Percentage of volatile matter in "dry coal."
(*i*) Moisture in coal as received.
(*j*) Whether or not bidder has absolute control over all of the coal from the mines upon which bid is based.

When the coal which the bidder proposes to furnish does not come from a mine on the Navy acceptable list, the bid will not be considered if the coal has an analysis indicating quality lower than the following:

(*a*) Moisture in "delivered coal," . . 3 per cent.
(*b*) Ash in "dry coal," . . . . 7 per cent.
(*c*) Volatile matter in "dry coal," . . 21 per cent.
(*d*) Sulphur in "dry coal," . . . 1¾ per cent.
(*e*) British thermal units per pound of "dry coal," 14,500,

If the analysis of the coal indicates quality equal to or better than this, the bidder may be awarded a contract for a small amount of coal, the contract being subject to cancellation if the coal proves unsatisfactory for use on board ship, or the amount called for being subject to increase if the coal proves satisfactory, in which case the mines supplying the coal would be placed on the Navy acceptable list and the coal would be classed as Navy standard steaming coal.

The following is the usual and most satisfactory method of determining the suitability of coal for naval use and the eligibility of the mine for the Navy acceptable list.

When the analysis of the coal indicates an acceptable quality the supplier is offered the opportunity of having the coal tested at the Naval Engineering Experiment Station at Annapolis, Maryland, the conditions being that the supplier deliver at the Experiment Station about twenty-five tons of coal, free of cost to the Government, and agree to defray the cost of the test, making a deposit of about three hundred and fifty dollars for this purpose. To insure that the coal tested

represents commercial deliveries of run-of-mine coal from the mine specified the sample is selected by a naval coal inspector or other Government representative.

If this test is satisfactory one cargo of coal is purchased from the supplier for use on board naval vessels and, if the ships using the coal report that it is satisfactory, the mine is placed on the Navy acceptable list, the coal is classed as Navy standard steaming coal, and the supplier has the same opportunity to secure awards of coal contracts as other suppliers of Navy standard steaming coal.

MOISTURE.

The percentage of moisture allowed is small because where coal is purchased on the weight basis the moisture is paid for as coal and is of no value as fuel.

All coal necessarily contains some moisture. It is probable that a small amount of moisture is harmless and possible that a limited amount of moisture is beneficial on account of the water vapor in the furnace facilitating the combustion of the volatile combustibles driven off from the coal. The addition of a small amount of moisture by wetting the coal before it is fired may result in a saving by preventing the fine coal from being blown up the smoke stack and by obtaining better combustion, but this gain may be offset by the increased amount of heat carried away by the water vapor.

Excessive moisture in coal results in a reduction in the capacity of the boiler, caused by a lower rate of combustion, and an increased loss of heat in the smoke-stack gases, due to a slightly higher flue-gas temperature, the amount of heat required to heat the moisture itself and the escape of a larger quantity of free hydrogen.

ASH.

The percentage of ash is limited because ash is an impurity of no value. The loss due to presence of ash in the coal depends as much upon the character as it does upon the quantity of the ash. The most important characteristic of Navy coal

is that it must produce as much power per pound of coal as possible, and, of course, a large percentage of ash reduces the heat value per pound of coal, reduces the steaming radius of the ship using the coal, and throws upon the firemen an excessive amount of work in handling the refuse.

If the ash is fusible it will form clinkers which may cling to the grate bars and reduce or entirely shut off the air from some parts of the grate and, in breaking the clinker from the grate, it may become mixed with coal and cause a large loss. Even if the ash is not fusible it covers the grate and increases the resistance of the fuel bed to the passage of air through it and causes a loss in cleaning the fires.

No reason can be definitely assigned as to why some coal forms clinkers in burning and others do not, even when the percentage of ash is greater. For example, George's Creek coal as a rule contains a higher percentage of ash than Pocahontas does, yet it forms almost no clinker, whereas Pocahontas forms a heavy clinker that is easily removed from the furnace.

VOLATILE MATTER.

The percentage of volatile matter is one of the most important characteristics of coal, because high volatile coal is not adapted for use in Naval boilers, and this fact is one that is hard to impress on many dealers, because a high volatile semi-bituminous coal may be very high in heat value and, judged purely on a heat unit basis, it would be ranked very high as fuel for any purpose.

The percentage of volatile matter is limited because the combustion space is restricted, the coal must be hand fired, and forced draft is necessary to produce the requisite rate of combustion for the development of full power.

Coals high in volatile matter can be burned efficiently where natural draft, and mechanical strokers giving a slow rate of heating to the coal, are used in connection with boilers having ample combustion spaces; but to produce a fairly good efficiency with high volatile coal with hand firing re-

quires far greater care and skill than is required with low volatile coal.

In hand firing the coal is heated through a range of about 2,400 degrees F. in two or three minutes. Experiments have shown that, with any given coal, the amount and quality of the combustible driven off the coal by heating depend largely upon the rate of heating, and that when the rate of heating the coal is rapid, as in hand firing, the total volatile matter driven off is not only high in quantity but contains much tar vapor and, of course, this condition is greatly aggravated when the coal is high in volatile matter.

The tar vapors and other heavy carbon-hydrogen compounds which are the products of distillation of coal, when heated rapidly, burn slowly and, in order to burn them completely, they must be kept a comparatively long time within the furnace. When burning coal in a Babcock & Wilcox boiler, as installed on the *Delaware*, for example, at the rate of 20 pounds of coal per square foot of grate surface per hour, with the fuel bed one foot thick, the time that each cubic foot of gas stays in the furnace is about .25 seconds, so that a large volume of smoke is unavoidably emitted, and, when the coal is high in volatile, this smoke is dense and black and contains a large amount of tar vapor and other carbon-hydrogen compounds which escape unburned with a resulting loss and the accompanying military disadvantage of a heavy cloud of smoke that is visible many miles.

When forced draft is used the rate of combustion is increased and the volume of volatile matter driven off greatly increased, so that the time available for the combustion of the products of distillation is greatly reduced and the speed of the gases through the tube space so increased that they reach the uptakes before their temperature is reduced below the point of ignition, so that combustion takes place in the uptakes and the smoke stack with consequent overheating and greatly accelerated deterioration of these parts.

Another disadvantage of high volatile coal is its liability to give off inflammable gases while stored in bunkers; and

while a high percentage of a volatile matter may not be a cause of spontaneous combustion of coal, it is a fact that nearly all cases of spontaneous combustion of coal in the Navy have occurred in coal that contained a fairly high percentage of volatile matter.

SULPHUR.

The amount of sulphur is limited because it is an undesirable element in coal and, as it usually occurs in combination as iron pyrites, it is worthless as fuel and, even in the free state, it has a heating value of only 4,000 B.t.u. per pound.

Sulphur has a decided effect upon the clinkering quality of the ash and, as a general rule, the tendency of a coal to clinker varies directly as the sulphur content and inversely as the ash content.

When coal containing a high percentage of sulphur forms clinkers that adhere to the grate, the sulphur attacks the grate bars and causes them to burn away and deteriorate rapidly.

The clinker may sometimes be prevented from running between the grate bars by blowing steam under the grate, this result being due to the fact that the steam is decomposed by the heat, and the cooling process of decomposition reduces the temperature of the clinker below its melting point.

The presence of sulphur in coal may be a contributary cause of spontaneous combustion, but opinion is fairly well divided as to whether or not sulphur is a cause of spontaneous combustion; however, inasmuch as it may be a cause, this point is mentioned as one reason for limiting the percentage of sulphur allowed.

HEAT VALUE.

The heating value of the coal is the most important quality because, in the purchase of coal, one is buying useful heat units, and in coal for the Navy it is necessary that the coal be high in heat units so fixed in the coal as to become available and productive when the coal is burned in the furnace of

a naval boiler because the vessels must be able to steam the greatest possible distance with the weight of fuel carried, and must be able to attain their maximum speed when it is required.

OXYGEN.

In the description of coal that might prove acceptable to the Navy no mention is made of oxygen, but a limit is placed upon the percentage of this element that is permissible by the specifications as to ash and heating value. The anticalorific value of oxygen in coal is practically equivalent to the anticalorific value of the same percentage of ash, so that even if the ash is low the heating value will be low also if the oxygen is high.

In addition to oxygen having the same anticalorific value as ash, its presence in coal is further objectionable because the oxygen content is an index of the avidity with which coal absorbs oxygen, high-oxygen coal absorbing oxygen more rapidly than low-oxygen coal. Spontaneous combustion of coal is due to the oxidation of coal in places where the heat formed cannot be dissipated, so that a coal which has a great avidity for oxygen is a coal that is liable to suffer from spontaneous combustion.

THE PURCHASE OF COAL ON A B.T.U. BASIS.

The Navy Department has been frequently urged to purchase coal under strict specifications that provide that the purchase price shall be a certain figure for coal that shows a heating value of a certain number of British thermal units per pound; that coal which falls below or exceeds in calorific value this number shall be purchased at a price less or greater than the base price by an amount depending upon the number of B.t.u. below or above the standard.

Theoretically, this is an ideal way of purchasing coal, but, practically, it is very unsatisfactory, because it frequently happens that coal which, in a laboratory test, would be rated very high on a B.t.u. basis, gives very poor results when

burned in the furnace of marine boilers, and conversely. In fact, one of the very best steaming coals on the market, and one which is largely used by sea-going steamers, would be barred from competition because it is lower in B.t.u.'s than other first-class coals and would, therefore, be forced to accept a lower price than such coals or abandon the Navy trade, which it probably would. This coal, George's Creek, has long been recognized by marine engineers as one of the most satisfactory on the market.

"The method of purchase adopted by the Navy is the result of experience in the actual burning of coal in marine boilers over a period of seventy years, and was not adopted without careful consideration of all the questions involved."

This method of purchasing coal would no doubt cause many disputes between the suppliers of coal and the Navy Department, and, in the end, would be unsatisfactory, because the Navy is not buying so many heat units but is buying high-grade coal that by actual use has been shown to be the best coal obtainable for use in Naval boilers, and the suitability of the coal for the duty required should be the controlling factor in the purchase of coal for use on board naval vessels.

WEST COAL COALS.

Practically all the coal for the Navy comes from the eastern fields which supply Pocahontas, New River, George's Creek and Eureka coal, but the Navy Department, for military and economic reasons, has endeavored to find a coal mined near the Pacific Coast that would be acceptable for the uses of the naval service.

Coal from British Columbia, Washington, Colorado and Utah, has been tested, but no coal has been found that is as satisfactory as the Navy standard steaming coals, and the investigations of the Bureau of Mines and of the Geological Survey fail to show any evidence of the existence of a western coal that would be satisfactory for naval purposes.

All western bituminous coals have a relatively high per-

centage of volatile matter. Many of the best have a large percentage of ash, and most of those tested on board naval vessels contained ash that formed a heavy clinker that adhered to the grate bars and bridge wall.

It has been said that Western coal would give better results if a different form of grate bar were used, but it is probable that the improvement, if any, resulting from the use of these bars would be slight; and there are serious objections to their use. Naval vessels may be required to proceed on short notice to places where Western coals are not obtainable. It would, therefore, be necessary for every vessel to carry two complete sets of grate bars—one for burning Western coals and the other for burning Eastern and Welsh coals. A set of the usual type of grate bars for the *Colorado* weighs, including ten per cent. of spares, about fifty tons. There would be no place to store these bars except in coal bunkers, where they would take the place of coal, and would still further reduce the coal-carrying capacity. The set of special bars would cost, with the necessary spares, about $3,000 for each ship of the *Colorado* class.

It is claimed that fairly good results would be obtained with Western coals if the firemen were especially trained to burn this coal. This is probably true, but it is not believed that it is possible, even with special grate bars, and with firemen skilled in using Western coal, to burn the best of these coals with results approaching those with Eastern coal. Besides, there are decided objections to training men to burn this coal to the exclusion of Eastern coal. In order to teach the men the most efficient method of burning the high volatile Western coals it would be necessary for them to forget what they now know about the use of Navy standard steaming coal, and it would result in reduced naval preparedness, because in time of war the best obtainable coal must be used and the men must be qualified to produce the maximum efficiency with this coal.

The existence of the Navy is justified only on the supposition that it will be kept at all times in the maximum state of

readiness for war. Wars often come with little, if any advance notice, and, if the Navy is allowed to fall into a state of unpreparedness in any detail with the expectation of perfecting that detail when war is threatened, it is inviting disaster. One of the most important of these details is the ability of the fleets to steam the greatest distance with the amount of coal they can carry, and to develop the maximum speed when required. This ability depends in a large degree upon the skill of the fireroom force, and this skill can be assured only by careful instruction and training. It is essential to the country's interest that the firemen on board its men-of-war should be kept as well trained as the gun crews. They are at present well trained. To make them unlearn what they now know, learn to handle a kind of coal which is to be used only in time of peace, and then depend upon their relearning, in time of threatened hostilities, to burn to best advantage the kind of coal that must be used in time of war, would be analogous to requiring the gun crews to use brown powder at target practice and depending upon them to learn, at the outbreak of war, to use smokeless powder, the only suitable kind in battle.

ALABAMA COAL.

There are coals in Alabama that are high in heat units and low in ash and sulphur, but all of them are relatively high in volatile matter. Several tests have been made on board ship with Alabama coal and the results were not unsatisfactory when burned in Scotch boilers under natural draft, but it is believed that the percentage of volatile matter is so great as to preclude the possibility of coal from any of the known mines proving satisfactory as Navy standard steaming coal.

ARKANSAS COAL.

There is coal in Arkansas whose analysis is such as to indicate that it would be satisfactory for naval use, and an evaporative test of this coal is to be made at the Naval En-

ginering Experiment Station in the near future. If this test should prove satisfactory it is possible that an additional source of supply for coal for naval use may be developed.

ALASKA COAL.

Under the direction of the Navy Department, tests have been made on board the U. S. S. *Maryland* of samples of coal from two fields in Alaska.

The tests of the Bering River coal in 1913 showed conclusively that coal from the mine (location) selected for test was entirely unsuitable for naval use.

The test of the Matanuska coal in 1914 showed that the coal represented by this sample is suitable in every respect for use in the naval service.

There are many kinds of coal in the Bering River field, and it is very probable that when this field is developed a coal will be produced that will be suitable for naval use. Coal from one entry in this field shows the following results on analysis:

Moisture (as received),	2.68 per cent.
Volatile matter (dry coal),	16.70 per cent.
Fixed carbon (dry coal),	78.40 per cent.
Ash (dry coal),	4.90 per cent.
B.t.u. per pound (dry coal),	14,962.

If there is coal in Alaska that is suitable for use in the naval service, this source of supply is protected, because the "Act to provide for the leasing of coal lands in Alaska, and for other purposes," approved October 20, 1914, carries the following provision:

"SEC. 2. That the President of the United States shall designate and reserve from use, location, sale, lease or disposition not exceeding five thousand one hundred and twenty acres of coal-bearing lands in the Bering River field and not exceeding seven thousand six hundred and eighty acres of coal-bearing

land in the Matanuska field, and not to exceed one-half of the other coal lands in Alaska: Provided, that the coal deposits in such reserved areas may be mined under the direction of the President when, in his opinion, the mining of such coal in such reserved areas, under the direction of the President, becomes necessary, by reason of an insufficient supply of coal at a reasonable price for the requirements of Government works, construction and operation of Government railroads, for the Navy, for national protection, or for relief from monopoly or oppressive conditions."

REPORT OF THE U. S. NAVAL RADIOTELEGRAPHIC LABORATORY.

L. W. AUSTIN, HEAD OF THE LABORATORY.

An account of the work of the laboratory from its beginning in 1908 to 1912 was presented to the Chief of the Bureau of Steam Engineering in 1912 and published in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XXIV, p. 122, 1912. The present report deals with the work which has been done since that time.

Most of the experimental work not involving large pieces of apparatus or high power has been done at the laboratory at the Bureau of Standards, and two chief electricians have been assigned as assistants in this laboratory and a large number of others, both chief electricians and civilian radio aids, have assisted in the experiments involving sending and receiving at a distance. At present there is but one laboratory assistant as the other has been employed for the last four months in connection with the operation of the Tuckerton Radio Station.

The work of the laboratory may be divided into three parts:

1. *Routine testing* of small apparatus offered to the Department for use in radio telegraphy. The following is a list of these tests:

Complete receivers	23
Condensers	32
Inductances	12
Detectors	16
Telephones	19
Wavemeters	8
Amplifones	11
Detector testers	4
Ammeters	6
Audibility boxes	15
Buzzers	1

2. *Instruction* of a limited number of naval electricians in the practice of high-frequency measurements. For this work a typewritten course of instruction has been prepared which it is expected will soon be revised and printed for the general use of the service. The number of men who have taken this training course up to the present is 67. This part of the work is at present discontinued on account of the lack of assistants.

3. *Scientific work,* involving the general experimental study of high-frequency phenomena. Since the last report the main scientific work of the laboratory has been along the following lines:

1. Energy losses in condensers (equivalent condenser resistance.)
2. The high-frequency resistance of inductances.
3. Antenna resistance.
4. Experiments in the efficiency of different antenna types.
5. Quantitative experiments on radiotelegraphic transmission.
6. The transmission efficiency of continuous and damped waves.
7. A study of the ultraudion circuits.
8. Prevention of atmospheric disturbances.

Articles on all the subjects either have been or are about to be published as noted in the list of publications at the close of this report. A brief resumé of these articles is given below.

1.—ENERGY LOSSES IN CONDENSERS.

This work was begun several years ago and an account of the general methods employed, was given in the report already cited. The most important results obtained since the last report have to do with the change of effective condenser resistance with the frequency. In this work the method employed was that of measuring a current in a high-frequency circuit, first with the unknown condenser in circuit and then with a variable air condenser of the same capacity. Fine wire resistance was then placed in series with the air condenser

until the current in the circuit was the same as that observed when the unknown condenser was in circuit. The series resistance then represented the equivalent resistance due to the dielectric losses in the condenser being tested. As the dielectric losses are in most cases independent of the voltage and as it was desired to keep free from brushing losses, the condenser circuit was excited by a buzzer-driven wavemeter.

TABLE I.*

Relation of Resistance and Wave Length in Glass Condenser.

Wave Length, meters.	Resistance, ohms.
385	0.7
650	1.3
910	2.0
1,325	3.0
1,905	4.8
2,360	5.4
3,100	7.5

Table I gives the equivalent resistance of a copper-coated glass-plate condenser of 0.00196 mf. capacity. If the results be plotted it will be seen that the resistance increases directly as the wave length. See Fig. 1.

2.—THE HIGH-FREQUENCY RESISTANCE OF INDUCTANCES.

While the high-frequency resistance of non-inductive resistances can be easily determined by substitution methods, the determination of high-frequency resistance of inductances offers great difficulties.

The principle of the method used is briefly the following: Two equal inductance coils are placed in identical oil calorimeters, one coil is heated by a high-frequency current and the

* Proc. Inst. of Radio Engineers, I, p. 35, 1913.

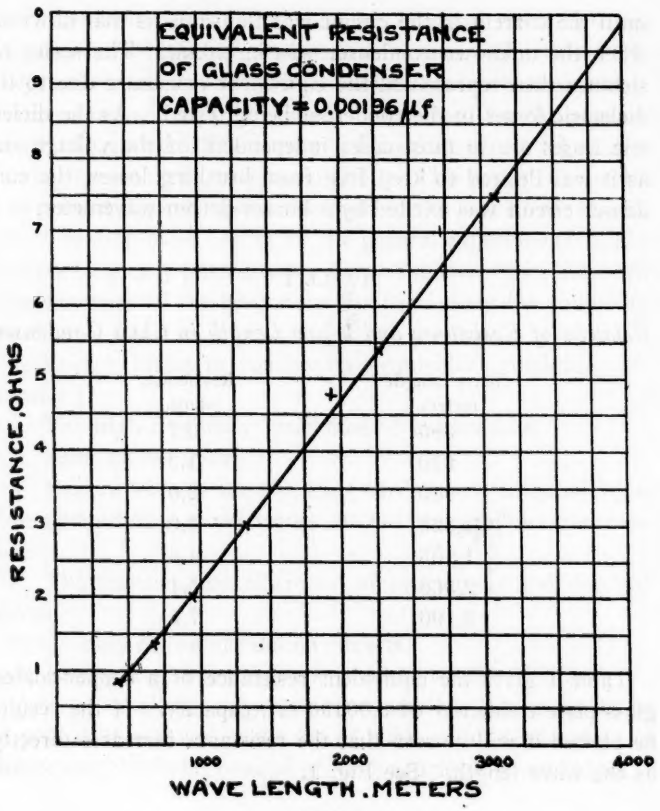


FIG. 1.

other by direct. When the calorimeters are both brought to the same temperature in equilibrium with their surroundings, the heat imparted to each per second must be the same. This heat is proportional to $I^2 R$, and the ratios of the resistances of the two coils at the given frequency and for constant current are inversely proportional to the squares of the currents. To compensate for the slight inequalities in the coils and calorimeters, the direct and high-frequency currents are interchanged and the mean values of the ratios of the current

squares taken. The high-frequency current is produced by a rotary spark gap in an oscillatory circuit coupled loosely to the circuit containing the inductance coil to be measured. The two circuits are brought to resonance at the frequency desired, and the high-frequency current through the inductance, regulated by varying the coupling. The current is read on a non-shunted hot-wire ammeter which has been accurately calibrated for high frequencies. The direct current for the other coil is supplied by a storage battery and the final regulation for equilibrium is made in this circuit. Equality of temperature between the two calorimeters is determined by a differential constant and copper thermoelement. The calorimeters are heated to from 20 degrees to 30 degrees above the temperature of the room. The uncertainty of the individual high-frequency current readings is approximately one part in fifty. The mean of thirty or more readings is taken as a basis for each calculation. The calorimeters, provided with motor-driven stirrers, are of glass, 15 cm. high and 10.8 cm. in diameter, and contain sufficient petroleum to cover the coils under experiment. The coils, of 0.40 mm. double silk-covered copper wire, are wound on glass, and the principal constants are given in Table II and their resistances in Table III.

TABLE II.

Diameter of double silk-covered copper wire = 0.04 cm.
Diameter of coils = 8.6 cm.
Turns of wire per centimeter = 18.9.

Coil.	Length. cm.	Turns of wire.	Inductance. m.h.
1	0.60	11.0	0.022
2	1.10	20.5	0.066
3	1.55	29.5	0.115
4	2.65	50.0	0.273
5	3.75	70.5	0.475
6	5.40	102.0	0.775

TABLE III.

Coil 1.		Coil 2.		Coil 3.		Coil 4.		Coil 5.		Coil 6.	
	R		R		R		R		R		R
m.	ohms.	m.	ohms.	m.	ohms.	m.	ohms.	m.	ohms.	m.	ohms.
360	1.45	550	2.06	780	2.87	780	6.60	1,440	7.75	1,500	13.3
550	1.09	780	1.70	970	2.54	970	5.93	1,900	6.55	2,000	11.8
780	0.86	970	1.51	1,150	2.20	1,440	4.65	2,400	6.28	2,500	10.9
970	0.72	1,150	1.40	1,440	2.02	1,900	4.03	2,900	5.93	3,000	10.2
D.C.	0.42	1,440	1.31	1,900	1.81	2,400	3.52	D.C.	2.68	3,500	9.6
...	...	D.C.	0.78	D.C.	1.12	D.C.	1.90	...	...	4,000	9.1
...	...	...	...	...	...	...	...	...	...	D.C.	3.88

After the determination of the resistance of the six pairs of standard coils a roller inductance of the Fessenden type was calibrated by comparison with the standard coils. The method used is as follows: A buzzer-driven wave meter is used to excite an oscillatory circuit containing a sensitive thermoelement and variable air condenser. By means of switches either one of the standard coils or the variable inductance can be inserted in this circuit, the deflections in each case being observed on the galvanometer of the thermoelement. Sufficient fine-wire resistance is then placed in series with the inductance giving the larger deflection and adjusted until its deflection is reduced to that of the other inductance. The resistance of the standard coil for the given frequency being known, the corresponding resistance of the variable inductance at this point is at once determined. When the variable inductance has been calibrated in this way for several points and at various wave lengths, it at once becomes a standard of comparison of resistance for any other inductances within its limits, by a method similar to the above. If the values of the resistances in Table III for any given wave length be plotted against the number of turns, that is, the length of wire in the different coils, it will be seen that the high-frequency resistance of the coils increases more rapidly than the number of turns. By plotting the resistance against the inductance,

however, the values will be found to be nearly in a straight line.

The increase in resistance of the comparatively fine wire in the above coils due to its being coiled up and not straight is only of the order of three to one. In larger wire the difference is much greater. For example, a fifty-turn coil 7.8 cm. long and 15.9 cm. in diameter wound with 1.29 mm. (No. 16) wire and having an inductance of 0.428 m.h., at a wave length of 1,000 meters has a resistance about seven times greater than that of the same wire straight at the same frequency, and twenty times greater than its direct-current resistance. From this it follows that it is difficult to reduce the resistance of coils below certain limiting values. For the wire sizes used in receiving sets this limit is about 16 ohms per millihenry at $\lambda = 1{,}000$ m. both for stranded* and solid wire. At $\lambda = 3{,}000$ m. the limit is, approximately, 9 ohms per millihenry for stranded wire or solid wire of more than 1 mm. in diameter. Sending inductances are slightly better, giving as a limit about 11 ohms at 1,000 m. and 7.5 ohms at 3,000 m.

3.—ANTENNA RESISTANCE.

A brief account of the early work on this subject was given in the first report. Since that time the resistance of a large number of antennae has been measured. Two characteristic antenna curves are given in Figure 2. That of the Bureau of Standards, which is typical of the resistance curves of all stations with poor grounds, shows a marked fall in resistance as the wave length is first increased, followed by a sharp rise with further increase in wave length. The curve of the U. S. S. *Maine* is typical of ships and land stations with thoroughly well-constructed ground systems. The falling part of the curve which is common to all stations represents the decrease in radiated energy as the wave length is increased. The radiated energy is expressed mathematically by

* The superiority of stranded wire does not show itself below $\lambda = 1{,}500$ m.

$$E = 160\pi^2 \left(\frac{h}{\lambda}\right)^2 I_s^2.$$

Where h represents the height to the center of capacity of the antenna, λ the wave length and I_s the current in the vertical portion of the antenna, the expression

$$160\pi^2 \left(\frac{h}{\lambda}\right)^2$$

is frequently spoken of as the radiation resistance, as it takes the same position in the energy equation as that occupied by R in the case of ohmic losses. It is particularly to be noted that the radiated energy at the sending station and the amount of

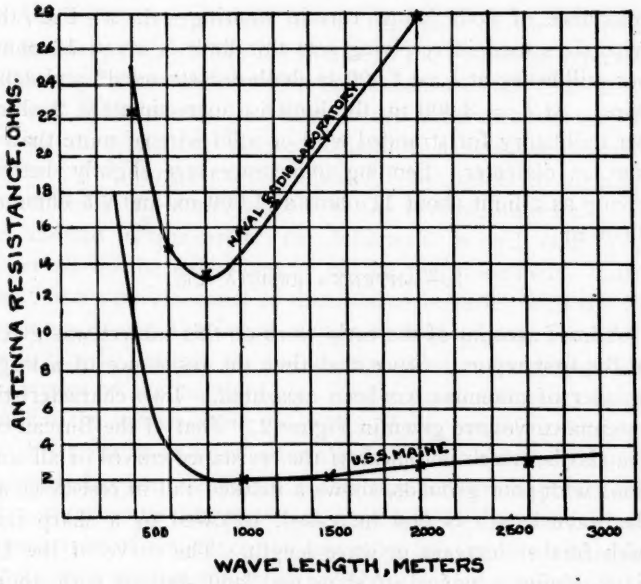


FIG. 2.

electro-motive force impressed on the antenna at the receiving station depend only on the height and shape of the antenna and the antenna current, the wave length remaining constant. The total antenna energy I^2 multiplied by (radiation resistance + ohemic resistance + earth resistance), which is often cited

in connection with wireless installations, has little to do with the actual sending effectiveness of the station, although total energy divided by the energy delivered to the spark gap has a technical interest as indicating the quality of the sending set.

Great difficulty was found at first in explaining the increase of resistance of most land stations with increasing wave length, as shown in the curve of the Bureau of Standards antenna, beyond 1,000 meters. The explanation was finally found in connection with the change of condenser resistance with changing wave length mentioned under condenser losses. Beyond a certain point the antenna resistance rises in proportion to the wave length just as does the condenser resistance in Table I. The connection between the two phenomena is evident when we remember that an antenna must be looked upon as a condenser, the antenna itself being the upper plate and the ground water being the lower. The greater part of the dielectric is, of course, air which is free from dielectric losses, but in most cases above the ground water there is a semi-conducting mass of earth which is a poor dielectric and undoubtedly furnishes the dielectric losses which produce the rise in the antenna resistance curve at the greater wave lengths. If the ground under the antenna be sufficiently well covered with wire this effect nearly disappears.

4.—EXPERIMENTS ON THE EFFICIENCY OF DIFFERENT ANTENNA TYPES.

In these experiments an attempt was made to gain information regarding the efficiency of different antenna types, using ordinary radiating wire grounds, such as are furnished with the portable sets, and various kinds of counterpoises. The antennas were supported on 40-foot portable masts erected on low, wet land near the Arlington station, thus ensuring good ground conditions.

The types of antennas used were: a triangle 85 feet on a side supported by three masts, a "T" 85 feet long, an "L" 85 feet long, an umbrella of six 85-foot wires, and an umbrella

of eight 40-foot wires. Each of the umbrellas was tried with its wires extended at 73 degrees to the vertical, and again at 45 degrees. In all the experiments the wave length was 1,000 m. and a current of approximately 2 amp. was maintained in the different antennas. Receiving measurements were made at the Radio Laboratory by means of thermoelements. Receiving experiments were also made on the antennas under test, the sending station being the Washington Navy Yard. The full data of the experiments are too extensive and involved to give here. The results may, however, be summarized as follows:

The triangle, the "T" and the "L" gave the best absolute results, both in sending and receiving, on account of their greater effective height. The 40-foot wire umbrella showed itself superior to the standard 85-foot type. The relative efficiency of the 40-foot wire umbrella, taking into account its lower effective height, is greater than that of any of the other antennas.

In some of the experiments with the umbrellas the ground net was replaced by a circular counterpoise of from two to six wires running around just outside the ends of the antenna wires, leaving the ground directly below the antenna bare. As the counterpoise was insulated, it was hoped to throw the electric field outward, thus producing a greater effective height. This object was to a certain extent attained in sending, the efficiency of the 40-foot wire antenna at 73 degrees being raised from 52 per cent. to 63 per cent.

An experiment was also made making use of a ground formed by driving pipes into the earth so as to get direct conductive connection with the ground water. The results, however, were found to be extremely poor compared with the net grounds and counterpoises.

In connection with this work it was desired to investigate the effect of metal towers used in supporting antennas. These were simulated electrically by extending wires from the top of the masts to the ground net. In the case of the triangle antenna it was found that the sending efficiency was reduced

20 per cent., and the receiving efficiency 11 per cent. by the presence of these artificial grounded towers.

In the experiments on directive effects it was found that it made no certain difference whether the "L" antenna was used in sending to the Radio Laboratory with its open end toward the Laboratory or away from it. Extending a counterpoise from the base of the "L" antenna 280 feet toward the Radio Laboratory was found to give a slightly improved received current. In the case of the 40-foot umbrella with antenna wires at 45 degrees a directive counterpoise was constructed extending 260 feet toward the Radio Laboratory, and another 240 feet away from the Laboratory. It was found that when the counterpoise toward the Laboratory was used as a ground, the received current was 342 microampères, but when the counterpoise away from the Laboratory was used, the received current was only 243 microampères. When connected to both counterpoises the received current was 295 microampères, the same as was observed with the net ground.

5.—QUANTITATIVE EXPERIMENTS IN RADIOTELEGRAPHIC TRANSMISSION.

The quantitative study of long-distance radiotelegraphic transmission was begun by the Navy Department in 1909–10 and continued in 1912 in connection with the testing of the high-power radio station at Arlington, Virginia. This station is equipped with a 100-kw. Fessenden rotary-gap sending set which gives an antenna current of approximately 100 ampères at a wave length of 3,800 meters. The aerial is triangular in shape and suspended between three steel towers, two of which are 450 feet in height while the third has a height of 600 feet. The capacity of the antenna is 0.01 m.f. and the height to the center of capacity 400 feet. Short range experiments showed that the effective height of the Arlington station was only about one-half the height to the center of capacity of the antenna. This appears to be generally true of land stations and is probably due to the fact that they are not

erected on sufficiently good conducting surfaces as assumed in the theory. The main scientific object of the experiments was the determination of the correctness of the Sommerfeld transmission formula for received current:

$$(1) \qquad I_R = 120\pi \frac{h_1 h_2 I_s}{\lambda d R} E^{-\frac{0.0019d}{\sqrt[3]{\lambda}}}.$$

where h_1 is the effective height of the sending antenna, h_2 the corresponding height of the receiving antenna, I_s the sending antenna current, λ the wave length, d the distance between the two stations, and R the effective high-frequency resistance of the receiving-antenna system.

The strength of the received signals was measured on the U. S. S. *Salem,* which made a voyage to Gibraltar and return for the carrying out of the tests. The total height of the *Salem's* antenna was 130 feet and the height of the center of capacity 114 feet. The effective high frequency resistance was 50 ohms at 3,800 meters. Signals were received by means of an electrolytic detector and their intensity was measured by the shunted telephone method which was described in the paper already cited. From the data thus obtained it was possible to determine the received antenna current I_R.

Table IV shows the results. Column five gives the experimental values as obtained from the smoothed curve of observations, and column three the values as calculated from the Sommerfeld formula (1). Column four gives the calculated values as obtained from a semi-empirical formula (2) made up of the first term of the theoretical formula but with the absorption term replaced by the absorption term which was found to be correct in the experiments made in 1910. The values in column four are seen to be in very fair agreement with the observed values.

$$(2) \qquad I_R = 120\pi \frac{h_1 h_2 I_s}{\lambda d R} E^{-\frac{0.0015d}{\sqrt{\lambda}}}.$$

The Sommerfeld theory takes no account of energy which may be brought to the receiving station by means of reflection or refraction in the upper atmosphere, and it is thought probable that it is this portion of the energy which produces the difference between the observed and theoretical results.

TABLE IV.

Arlington Received on the "Salem," February–March, 1913.

Resistance.		Received current 10^{-6} ampères.		
		Calculated.		
Miles.	Km.	Eq. 1.	Eq. 2.	Obs.
300	556	335.000	431.00	410.0
400	740	200.000	278.00	300.0
500	925	128.000	195.00	225.0
600	1,110	85.200	140.00	160.0
800	1,480	40.700	79.00	95.0
1,000	1,850	20.700	47.60	59.0
1,200	2,220	11.000	29.70	34.0
1,500	2,780	4.420	15.30	19.0
2,000	3,700	1.070	5.65	5.0
2,500	4,630	0.280	2.20	.....
3,000	5,560	0.074	0.84	.....

6.—THE TRANSMISSION EFFICIENCY OF CONTINUOUS AND DAMPED WAVES.

An extensive series of experiments have been carried on by the Laboratory on the transmission of damped and undamped waves. It was found that for distances up to 1,000 miles the received antenna currents were practically the same in the two cases for the same sending antenna current. During the voyage of the *Salem* comparison was made between the received day signals from the Fessenden spark set and those from a Poulsen arc temporarily installed at Arlington. For distances above 1,200 miles the arc seemed to be superior for the same sending current. These results were verified later

by receiving the two types of radiation at the naval radio station at Colon. More recently observations have been made on the signals from the 200-kw. spark set at Nauen, Germany, and the 200-kw. high-frequency machine installed in the same station. The wave length in both cases is 9,400 meters. The current with the machine probably varies from 130 to 150 ampères, while the spark antenna currents probably are somewhat higher. At Tuckerton, N. J., observations show that the continuous oscillations are very much stronger, while at the Naval Radio Laboratory the spark signals have never been heard while the machine signals vary between 100 and 5,000 audibility.

7.—A STUDY OF THE ULTRAUDION CIRCUITS.

During the past year much of the time of the Laboratory has been devoted to the development of the DeForest ultraudion. This instrument is a modification of the earlier type

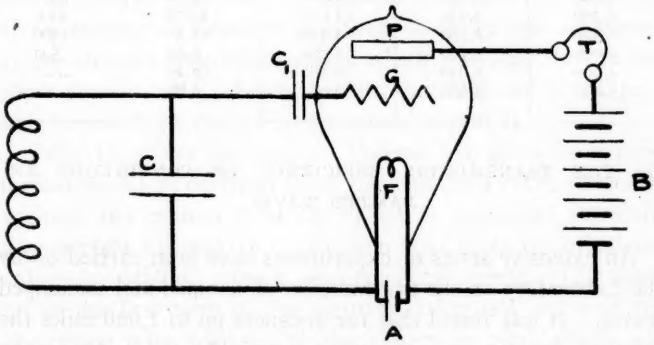


FIG. 3.

of DeForest audion detector, and differs from it only in the method of connecting the oscillatory circuit to the detector. Figure 3 shows the old or filament connection, while Figure 4 shows the ultraudion or plate connection. Under proper conditions of capacity, inductance, resistance, etc., continuous

oscillations* are set up in the circuit connected to the ultraudion in Figure 4. If these oscillations do not take place the

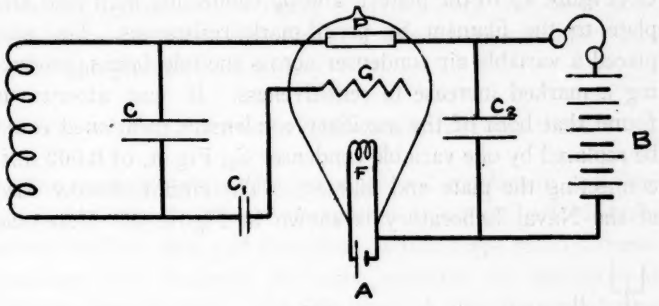


FIG. 4.

ultraudion behaves much like an ordinary audion except that it is many times more sensitive. Undamped oscillations are not heard and spark signals produce the natural note of the sending spark. If oscillations are produced by the audion in the connected circuit, by slightly detuning this circuit from the incoming continuous oscillations, musical beats are produced between the local and incoming oscillations, as in the case of the Fessenden heterodyne. The pitch of the beat tone is entirely determined by the degree of detuning. If the audion is allowed to oscillate when receiving spark signals regular beats cannot be produced and the spark note is roughened.

With the simple circuit without the condenser C_2, Figure 4, oscillations in general do not take place with the capacity C greater than about 0.0005 m.f. and the oscillations are very easily destroyed if any considerable amount of energy is taken from the circuit or if strong atmospheric disturbances impinge upon it.

The main improvements have been along the line of giving greater stability to the oscillations and increasing the sensitiveness. It has been found by Chief Electrician Eaton, assistant

*The presence of these oscillations can be detected by touching with the finger any part of the oscillatory circuit on the grid side of the inductance. If oscillations are present a noise will be heard in the telephones.

in the Laboratory, that the stability of oscillations can be much increased by placing a small condenser across from the grid G (Figure 4) to the plate P and by connecting both grid and plate to the filament by pencil-mark resistances. He also placed a variable air condenser across the telephones, producing a marked increase in sensitiveness. It was afterwards found that both of the auxiliary condensers mentioned could be replaced by one variable condenser C_2, Fig. 4, of 0.002 m.f. connecting the plate and filament. The circuit as now used at the Naval Laboratory is shown in Figure 5. Here the

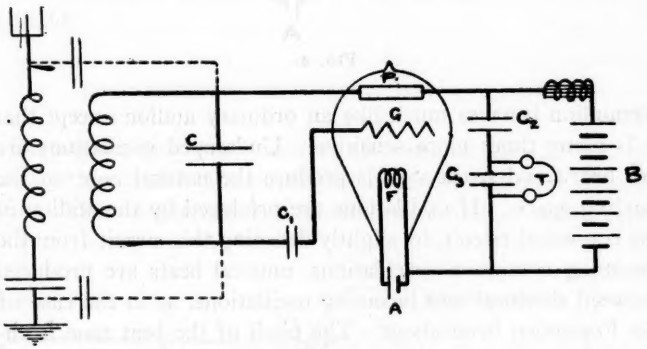


FIG. 5.

original telephones of Figure 4 are replaced by a choke coil and the telephones placed in the shunt circuit P F. The condenser C_2 should not be less in capacity than 0.01 m.f. for full strength of signal, and the condenser C_3 is variable from 0 to 0.002 m.f. By making C_2 variable certain types of atmospherics can be much reduced by decreasing its capacity. With this form of circuit it is possible to use a capacity of somewhat more than 0.002 m.f. for the tuning condenser C of the secondary circuit. The best results are obtained, however, with a capacity not greater than 0.001 m.f.

The stability of the oscillations can be increased and the sensitiveness improved by making use of a static coupling of about 0.0001 m.f. between the primary and secondary cir-

cuits, in connection with the magnetic coupling. The condenser connections for the static coupling are shown by dotted lines in Figure 5. For many purposes only the lower condenser, the one connecting the grid side of the secondary circuit to the ground, is necessary.

8.—PREVENTION OF ATMOSPHERIC DISTURBANCES.

The most important practical unsolved problem in radio reception is undoubtedly the prevention of atmospheric disturbances, for it is this, and this alone, which keeps radio communication from attaining the same certainty of operation as that attained by wire. For this reason a vast amount of time has been spent in the Laboratory in testing every device which has been proposed by radio experimenters or which has been thought of by the Laboratory staff. While most of these devices have proved useless, a few have given some degree of success.

One of the best methods so far tried in the case of ordinary receiving sets with contact detectors is that of placing a contact composed of silicon and arsenic around the receiving antenna inductance. Contacts of this character have the property of offering a very high resistance to small electromotive forces, and a comparatively low resistance to electromotive forces of any considerable magnitude. In this way strong atmospheric disturbances and powerful interfering signals may be partly shunted to earth, while the weaker received signals pass through the inductance with practically no diminution in intensity. The general character of the action is shown in Table V, the deflections being those of a galvanometer attached to the regular receiving detector, with the disturbance-preventing circuit connected and disconnected. The antenna was excited by a tuned buzzer circuit with different degrees of coupling. It is seen that the weak signals such as ordinarily are received are not at all weakened by the disturbance preventer, while the strong signals are cut down to less than a tenth of their full value.

TABLE V.

Disturbance Preventer.

In. Deflection. mm.	Out. Deflection. mm.
0.5	0.5
1.5	1.5
3.5	4.5
7.0	22.0
15.0	130.0
21.0	240.0
40.0	off scale
2.5	2.5

This device has been used to a considerable extent in receiving in various radio stations. Its chief practical drawback seems to be the difficulty of getting correct adjustment of the contact, this being considerably more critical than the ordinary detector adjustment. It is also found that in the case of very large antennas the atmospheric strokes are frequently so powerful that the disturbance preventer contact is broken down, thus limiting the applicability of the device in large stations.

For the best results the antenna should contain considerable inductance with a series condenser if necessary. The coupling should be loose.

In the new audion circuits the prevention of atmospheric disturbances is a somewhat different problem from that met in the old-type circuits. The use of loose coupling is less advantageous and appears to result in a diminution in strength of signal quite as rapid as the diminution of the atmospherics.

It is extremely fortunate that the oscillating audion is far less sensitive to disturbances than to the signals, but nevertheless atmospherics can be extremely troublesome and thus far no means have been found for controlling them. With certain types of disturbance considerable relief can be obtained

by reducing the capacity of the condenser C_2 in Figure 5. The connection of the telephones through a tuned variable coupling iron core telephone transformer, as shown in Figure 6, gives further improvement. In this the primary and secondary may have approximately the same inductance: *i. e.*, about 200 milli-

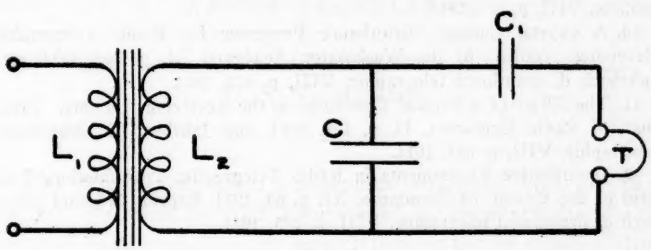


FIG. 6.

henries, while C is about 0.01 m.f. and C_1 is 0.01 or 0.02 m.f. By making both C and C_1 variable the transformer can be tuned to any tone desired. The primary of the transformer is inserted in the place of the telephones as shown in Figure 3.

Below is given a list of the papers published by the Laboratory since the last report:

1. Antenna Resistance. Bulletin of the Bureau of Standards, IX, p. 65, 1912. Reprint 189, and Jahrbuch d. drahtlosen telegraphie, V, p. 574, 1912.

2. Energy Losses in Some Condensers Used in High-Frequency Circuits. Bulletin of the Bureau of Standards, IX, p. 73, 1912. Reprint 190, and Jahrbuch d. drahtlosen telegraphie, VII, p. 222, 1913.

3. Comparison of Damped and Undamped Oscillations. Journal of the Washington Academy, II, p. 111, 1912, and Jahrbuch d. drahtlosen telegraphie, V, p. 524, 1912.

4. Suitable Wire Sizes for High-Frequency Resistance. Journal of the Washington Academy, II, p. 112, 1912, and Jahrbuch d. drahtlosen telegraphie, VI, p. 588, 1912.

5. The Relation Between Effective Resistance and Frequency in Radio Telegraphic Condensers. Proc. Institute of Radio Engineers, I, p. 35, 1913.

6. The High-Frequency Resistance of Inductances, Journal of the Washington Academy, III, p. 94, 1913, and Jahrbuch d. drahtlosen telegraphie, VIII, p. 159, 1914.

7. The Measurement of Received Radiotelegraphic Signals. Journal

of the Washington Academy, III, p. 133, 1913, and Jahrbuch d. drahtlosen telegraphie, VII, p. 628, 1913.

8. A Comparison of Arc and Spark Sending Apparatus. Journal of the Washington Academy, III, p. 284, 1913, and Jahrbuch d. drahtlosen telegraphie, VII, p. 506, 1913.

9. Difference in Strength of Day and Night Signals. Journal of the Washington Academy, III, p. 326, 1913, and Jahrbuch d. drahtlosen telegraphie, VIII, p. 381, 1914.

10. A Crystal Contact Disturbance Preventer for Radio Telegraphic Receiving. Journal of the Washington Academy, III, p. 386, 1913, and Jahrbuch d. drahtlosen telegraphie, VIII, p. 481, 1914.

11. The Effect of a Parallel Condenser in the Receiving Antenna. Pros. Inst. of Radio Engineers, II, p. 131, 1914, and Jahrbuch d. drahtlosen telegraphie, VIII, p. 524, 1914.

12. Quantitative Experiments in Radio Telegraphic Transmission, Bulletin of the Bureau of Standards, XI, p. 69, 1914, Reprint 226, and Jahrbuch d. drahtlosen telegraphie, VIII, p. 575, 1914.

POSSIBLE APPLICATION OF THE DRZEWIECKI METHOD TO THE DESIGN OF WATER PROPELLERS.

By Assistant Naval Constructor H. E. Rossell, U. S. N., Member.

The remarkably high efficiencies obtained with air propellers designed by the Drzewiecki or constant incidence method have caused much speculation as to the applicability of this method of design to water propellers. The best air propellers at present have efficiencies as high as 85 per cent., while the highest obtained with water propellers is about 75 per cent.

As is well known, the conventional methods of designing water propellers all consider the propeller to be analogous to a nut working on a thread. The distance the propeller would move along its axis in one revolution, if it were working on a rigid thread, is called the true pitch. However, in one revolution, the propeller actually moves through the water in which it is working an axial distance less than the true pitch. This distance is ordinarily called the effective pitch, the percentage difference between true and effective pitch being called the real slip. The speed of the wake of the ship, in which the propeller works, is assumed to be some percentage of the speed of the ship. In designing a propeller, the speed of advance through the water, the thrust which must be developed, and the revolutions are usually fixed. The diameter can generally lie within certain rather narrow limits. The design variables are, therefore, the pitch and the blade width. Chief Constructor Taylor's method is based on the performance of model propellers, as is Professor Peabody's, while Captain Dyson's method is based on the performance of full-size pro-

pellers. Any of these methods will give a propeller which will do the work at an efficiency of from 65 to 70 per cent. With few exceptions all water propellers have a helicoidal after surface, the thickness all being on the forward side or back of the blades. A plane section of a blade, parallel to the axis of a propeller, usually has its maximum ordinate at the center of the chord and is symmetrical with respect to this ordinate.

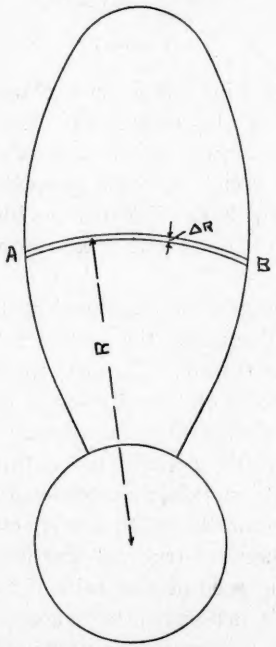


FIG. 1.

In the Drzewiecki method of design any section of the blade, as AB, Fig. 1, is considered as a section of a wing, having a velocity V_t through the medium in which it is working. V_t is the velocity compounded from the speed of rotation at radius R, and the speed of advance through the water. The curvature of the section, AB, is disregarded, and it is assumed

to act as if it were a plane section moving at a velocity, V_t, with respect to the air, or water. The forces, acting on the section, may be resolved into lift component, $\triangleleft L$, and drift component, $\triangleleft D$, respectively perpendicular to and along the direction of motion. The angle, cot $= {}^1\triangleleft L \div \triangleleft D$, we will call G; and the angle between the direction of motion and

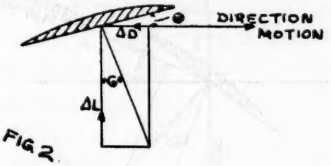


FIG 2.

the chord of the section we will call θ, Fig. 2. Angle A, Fig. 3, is the angle between the direction of motion and the normal to the axis of the propeller.

Force along axis $= \triangleleft L \cos A - \triangleleft D \sin A$.
Force perpendicular to axis $= \triangleleft L \sin A + \triangleleft D \cos A$.

$$\text{Efficiency of element} = \frac{(\triangleleft L \cos A - \triangleleft D \sin A) \sin A}{(\triangleleft L \sin A + \triangleleft D \cos A) \cos A}$$

$$= \frac{(\cos G \cos A - \sin G \sin A) \sin A}{(\cos G \sin A + \sin G \cos A) \cos A}$$

$$= \frac{\cot (A + G)}{\cot A}$$

$\cot A = 2\pi R \div P_e$, where P_e is the effective pitch.

Now, if the lift per unit area and the ratio of lift to drift at each section are known, the forces acting along the axis at all parts of the blade are known, and can be summed up to get the thrust.* A large number of tests have been made in air on model wings of various sections. These tests usually give the ratio of lift to drift and the lift, either in the form of a coefficient or directly, for various angles of incidence. By

* For an excellent description of the method used in actually making the design, see Report No. 65, by Mervyn O'Gorman, Technical Report of the Advisory Committee for Aeronautics, 1911-1912.

angle of incidence is meant the angle between the chord of the wing and the direction of motion. It is the angle θ, in Fig. 3. Fig. 5 shows the lift ÷ drift and lift coefficient curves

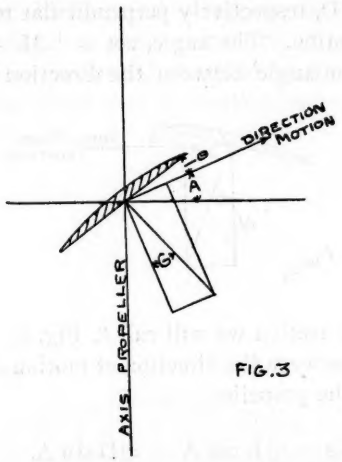


FIG.3

for two wings, selected from the Technical Report of the Advisory Committee on Aeronautics, London, 1912–1913. These curves show that each wing has a definite angle of incidence at which the ratio of lift to drift is a maximum. For wing No. 0 in Fig. 5 this incidence is 8 degrees; while for wing No. 2 it is 4 degrees. Maximum ratio of lift to drift gives a minimum gliding angle G, and maximum efficiency for any section of the propeller having the most advantageous angle of incidence. It is, therefore, the aim, in this method of design, to give the propeller blade such a form as to cause each section to operate at the angle of incidence for which the ratio of lift to drift is a maximum. In other words, the angle θ, Fig. 3, is kept the same for all sections of the blade.

In selecting a suitable section for a propeller the character of the lift curve is of importance. This section adopted must have a good lift at the incidence used in order to get the desired thrust with reasonable blade width and propeller diameter.

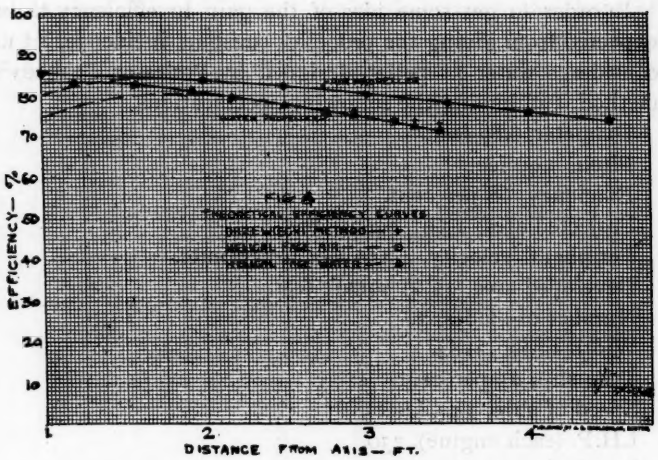
In order to get some idea of the gain in efficiency to be expected from giving the propeller constant incidence, let us consider a water propeller, designed by Professor Peabody's method, and having the following specifications.

Diameter, 6.87 feet.
Pitch, 7.90 feet.
Efficiency, 68 per cent.
Projected area, 10.01 square feet.
Peripheral velocity, 4,854 feet per minute.
Developed area, 12.21 square feet.
Number of blades, 3.
Width ratio, .2184.
True slip, 25 per cent.
Speed of ship, 14.5 knots.
I.H.P. (each engine), 730.
Revolutions, 225.
Wake factor (assumed), .1.
Speed of wake $(.9 \times 14.5 \times 6{,}080) \div 60 = 1{,}322$ feet per minute.
Effective pitch, $P_e = 1{,}322 \div 225 = 5.877$ feet.
$\cot (A - \theta) = 2\pi R \div P$, where P is the true pitch.

Solving for θ, we find that it varies from 9.2 degrees at radius of 10 inches to 4.9 degrees at the blade tips. For the sake of comparison, let us assume that the section of the propeller is that of wing No. 0 in Fig. 5, and that the curve of lift ÷ drift is the same in water as in air. The ratio of lift to drift may be taken from the curve for each angle of incidence, and the theoretical efficiency curve for the helical propeller worked out. This curve is shown in Fig. 4.

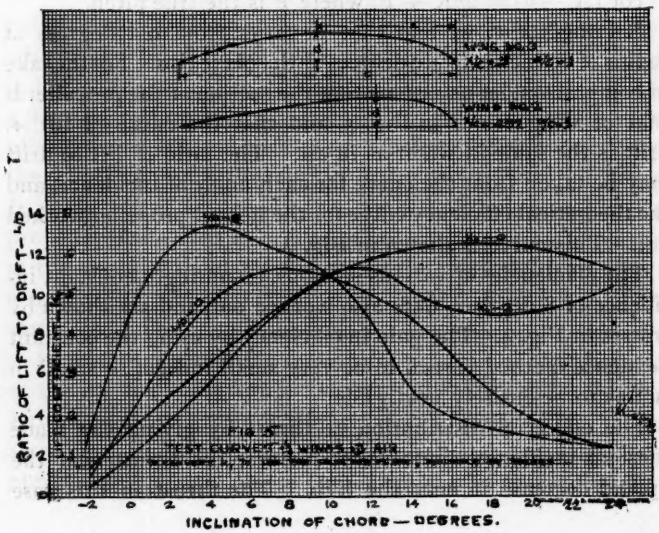
The efficiencies at various sections, assuming a constant incidence of 8 degrees, were also worked out, and are shown by crosses in Fig. 4. The gain in efficiency by using the constant incidence method is seen to be very small indeed, so small in fact as to be entirely negligible.

The effect on distribution of thrust by giving the blade constant incidence would be to increase the thrust near the tip and decrease it near the hub. The thrust in this case



would be increased at the blade tip about 10 per cent. This would probably cause a greater tendency toward cavitation.

Now suppose that we have adopted section No. 2 for an air propeller with the following specifications:



Diameter, 9 feet.
Slip, 25 per cent.
Revolutions, 1,200.
Speed, 90 miles per hour.

The theoretical efficiency curve assuming a helicoidal surface is shown in Fig. 4. The efficiencies for constant incidence are, as before, designated by crosses. Actually the high efficiency cannot be maintained near the hub, where the sections must be thickened in order to get the necessary strength. The probable drooping of the efficiency curves is shown in dotted lines in Fig. 4. The gain in efficiency by using the constant incidence method of design is seen to be even smaller than in the case of the water propeller.

CONCLUSION.

The Drzewiecki method is well adapted to the design of air propellers. By using it and the published tests on wing sections, one can design a propeller for any air craft, and be confident of getting very high efficiency and of developing the required torque at the designed revolutions. There is, however, no ground for the belief that the high efficiency is obtained by using constant incidence. The gain from this source is of the order of .2 of one per cent. The great advance made in air propeller efficiency is attributable to the more efficient sections which have been discovered.

There seems to be no advantage in applying this method to the design of water propellers. The helical propeller is cheaper to make of metal than the Drzewiecki propeller. The conventional methods of design are much easier to use than the constant incidence method, and give as good efficiencies. Moreover there are no tests available of wings in a current of water. The tests of wings in air cannot be used with any certainty, because the frictional and eddy-current losses are probably much larger in water than in air. The method does suggest a very rational way of investigating the efficiencies

of various sections which might be used for water propeller blades. It might, for instance, be much easier to test a given section for lift and drift in water than to determine, by test, the efficiency of a propeller built with this section. It is now a recognized fact that the form of the back of the blade section has considerable influence on the performance of either an air or water propeller. If any marked advance is to be made in the efficiencies of water propellers it must be from the discovery of a more efficient blade section over those which have been tried heretofore.

U. S. S. *BALCH.*

CONTRACT—TRIAL PERFORMANCE.

By Henderson B. Gregory, Associate.

Torpedo-boat destroyer No. 50, the *Balch,* is one of eight destroyers authorized by an Act of Congress, approved March 4, 1911. She is a sister ship of the *Aylwin, Parker* and *Benham,* the four vessels being built under contract by the William Cramp and Sons Ship and Engine Building Company of Philadelphia, Pa., at a price of $756,100.00 each. The time of completion for the *Balch* was twenty-four months from the date of signing the contract, September 7, 1911. She was designed for a speed of 29.5 knots, at about 1,036 tons displacement, with the main engines developing 16,000 shaft horsepower.

The *Balch* is a twin-screw vessel fitted with a combination of Cramp-Zoelly turbines and reciprocating engines, one of each on each shaft. For speeds of 17 knots and above the turbines alone are used, and for the lower cruising speeds the reciprocating engines are used in combination with the turbines, disconnecting couplings being installed for readily throwing the latter in and out of gear. Steam is supplied by four oil-burning White-Forster water-tube boilers, arranged in pairs in two separate compartments.

The hull and machinery are duplicates of the *Aylwin, Parker* and *Benham,* a complete description of which was published in the February, 1914, number of the Journal of the American Society of Naval Engineers, and which likewise applies to this vessel.

PRINCIPAL HULL AND MACHINERY DATA.

Hull dimensions:	
Length on L.W.L., feet and inches	300–0
over all, feet and inches	305–3
Breadth at L.W.L., feet and inches	30–4
Ratio, breadth to length (L.W.L.)	10.064
Draught to L.W.L., feet and inches	9–2½
Displacement corresponding, tons	1,010
per inch at L.W.L., tons	14.21
Area immersed midship section, square feet	190
Coefficient of fineness, block	0.415
Main turbines:	
Working pressure at chest, pounds gage	240
R.P.M., designed	620
S.H.P., designed	16,000
Number of stages, ahead	16
astern	8
Cruising engine:	
Type	Vertical, inverted, direct-acting, compound, jacketed.
Cylinder, H.P., diameter, inches	13
L.P., diameter, inches	25
Stroke, inches	12
Crank angle, degrees	180
R.P.M. (corresponding to a speed between 15½ and 16 knots)	300
I.H.P. (corresponding to a speed between 15½ and 16 knots)	460
Boilers:	
Type	White-Forster, oil burning, water tube.
Working pressure	260
Number	4
Furnace volume, each, cubic feet	452
Number of smoke pipes	4
Area through each smoke pipe, square feet	16.57
$\frac{\text{Area through smoke pipe}}{\text{Furnace volume}}$	0.03666
Heating surface, each, square feet	5,400
$\frac{\text{Heating surface}}{\text{Furnace volume}}$	11.9
Oil burners, number each boiler	11
type	Schutte-Koerting.
Diameter, inches, tubes (outside)	1
downcomers (inside)	8⅛
steam drum (inside)	48
lower drums (inside)	19½

TRIALS.

The following trials were required by the contract:

(*a*) A progressive trial over a measured-mile course for standardizing the screws, extending from maximum speed down to a speed of twelve knots.

(*b*) A full-speed trial of four hours' duration in the open sea, in deep water, at the highest speed attainable, the average speed to be not less than 29.5 knots, with the average air pressure in the firerooms not exceeding 6 inches of water and the steam pressure at the high-pressure turbine not to exceed 240 pounds above the atmosphere.

(*c*) A fuel-oil and water-consumption trial of four hours' duration in the open sea at an average uniform speed of 24 knots as nearly as possible.

(*d*) A fuel-oil and water-consumption trial of four hours' duration in the open sea at an average speed of 15½ knots as nearly as possible. This trial to be made as nearly as possible under service conditions, with the cruising engines connected up and in use.

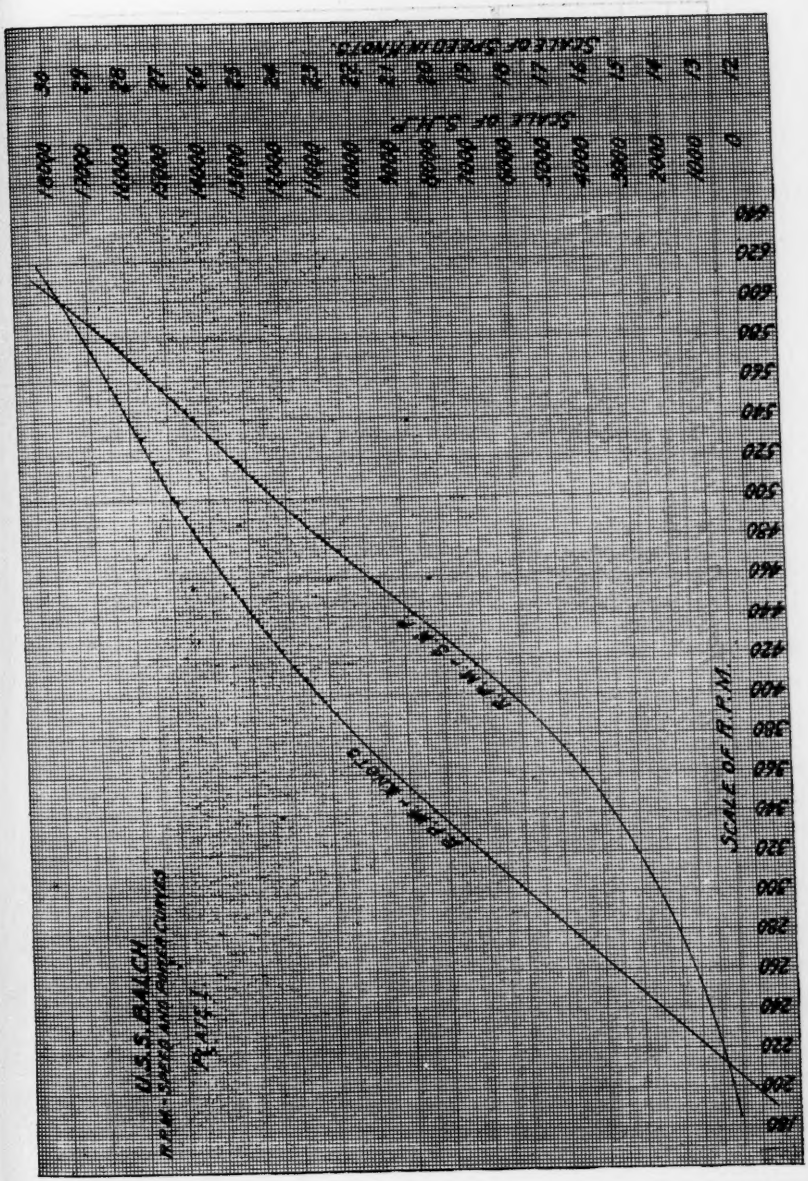
(*e*) An endurance trial of twenty hours' duration in the open sea at an average speed of 15½ knots as nearly as possible, under service cruising conditions, with cruising engines connected up and in use.

(*f*) A fuel-oil and water-consumption trial of four hours' duration in the open sea with the cruising engines connected up and in use, at an average speed of 12 knots as nearly as possible.

Fuel Oil Consumption Guarantees.—The contractors guaranteed that the fuel-oil consumption per knot run for all purposes, including that necessary for all auxiliaries in use on the trials, would not exceed 705 pounds at the guaranteed maximum speed of 29.5 knots, 445 pounds at 24 knots, 213 pounds at 15½ knots, and 170 pounds at 12 knots. For the purpose of ascertaining the fuel-oil consumption the fuel oil burned on trials (*b*), (*c*), (*d*) and (*f*) was carefully measured, from which the consumption at the stipulated speeds was computed

TABLE I - STANDARDIZATION TRIAL DATA - FEB. 21 & 22, 1914 - U.S.S. BALCH.

Nº of Run	Time on Course		Speed in Knots	Revs. to go One Knot			R.P.M.			S.H.P.		
	Mins.	Secs.		Star. Engine	Port Engine	Mean	Star. Engine	Port Engine	Mean	Star. Engine	Port Engine	Total
1	4	11.1	14.337	1042.4	1045.8	1044.1	248.75	249.56	249.15	751	681	1432
2	3	50.3	15.632	961.6	959.8	960.7	250.22	249.75	249.99	771	697	1468
3	4	4.3	14.736	1009.7	1010.2	1000.0	248.00	248.10	248.05	734	705	1439
MEAN OF GROUP			15.084						249.30			1452
4	3	14.3	18.528	1013.3	1015.3	1014.3	312.94	313.56	313.25	1418	1380	2798
5	3	12.7	18.682	1002.5	997.4	1000.0	312.05	310.46	311.25	1429	1385	2814
6	3	12.3	18.721	998.0	1000.9	999.5	311.39	312.29	311.84	1445	1358	2803
MEAN OF GROUP			18.654						311.90			2807
7	2	45.7	21.726	1009.9	1011.8	1010.9	365.23	366.23	365.73	2373	2205	4578
8	2	46.2	21.661	1021.5	1026.6	1024.1	368.84	370.66	369.75	2434	2331	4765
9	2	43.1	22.072	999.9	1006.5	1003.2	367.73	370.18	368.95	2405	2373	4778
MEAN OF GROUP			21.780						368.35			4722
10	2	32.0	23.684	1062.8	1060.9	1061.9	419.15	418.42	418.78	3818	3611	7429
11	2	27.0	24.490	1025.6	1025.5	1025.6	418.27	418.23	418.25	3693	3517	7210
12	2	32.4	23.622	1071.7	1070.0	1070.9	422.10	421.43	421.76	3871	3734	7605
MEAN OF GROUP			24.072						419.26			7364
13	2	11.1	27.460	1089.1	1085.8	1087.5	498.22	496.71	497.46	5989	5812	11801
14	2	17.4	26.201	1141.8	1142.2	1142.0	498.60	498.76	498.68	6327	6085	12412
15	2	9.8	27.735	1083.7	1075.8	1079.8	500.77	497.12	498.94	6159	6095	12254
MEAN OF GROUP			26.900						498.44			12220
19	4	35.5	13.067	885.4	882.9	884.2	193.53	192.99	193.26	358	322	680
20	5	32.8	10.817	1058.7	1074.0	1066.4	190.98	193.74	192.36	321	335	656
21	4	51.8	12.337	935.6	935.5	935.6	191.97	191.95	191.96	344	299	643
MEAN OF GROUP			11.760						192.50			659
23	2	2.9	29.292	1204.7	1197.4	1201.1	588.37	584.81	586.59	8455	—	—
24	2	2.5	29.388	1189.3	1188.1	1188.7	583.28	582.69	582.98	8382	8472	16854
25	2	5.2	28.754	1220.0	1206.3	1213.2	584.41	577.85	581.13	8363	8078	16441
MEAN OF GROUP			29.206						583.42			16648
26	1	59.6	30.100	1189.1	1180.3	1184.7	597.52	593.07	595.29	8784	8632	17407
27	2	4.4	28.939	1257.4	1244.4	1250.9	606.27	599.98	603.12	8912	9024	17936
28	1	57.0	30.769	1175.2	1166.5	1170.9	603.76	599.26	601.51	9014	8749	17763
29	2	5.6	28.662	1276.3	1261.0	1268.7	609.94	602.60	606.27	9135	8931	18066
30	1	54.5	31.441	1151.8	1146.9	1149.4	603.80	601.26	602.53	9015	8941	17956
MEAN OF GROUP			29.786						602.46			17862


U.S.S. BALCH
H.P.M. SPEED AND POWER CURVES
PLATE I.
H.P.M.-KNOTS
H.P.M.-S.H.P.
SCALE OF R.P.M.
SCALE OF S.H.P.
SCALE OF SPEED IN KNOTS.

| | | |

TABLE II – TRIAL DATA – U.S.S. BALCH.

	4-Hrs.' Full Power Trial (b)	4-Hrs.' 24-Kt. Trial. (c)	4-Hrs.' 15½-Kt. Trial. (d)	4-Hrs.' 12-Kt. Trial. (f)
Date of Trial	Feb. 26, 1914.	Feb. 22, 1914	Feb. 27, 1914.	Feb. 23, 1914
Speed in Knots	29.618	24.031	15.594	12.208
Draught, mean on trial, feet and inches	9 - 4 ⅜	9 - 6½	9 - 5 ⅜	9 - 7 ⅜
Displacement, corresponding, tons	1023.9	1053.0	1050	1065.5
Slip of Propellers, percent, Starb'd	24.59	14.51	7.87	7.06
Port	24.62	12.83	8.63	7.65
Mean	24.605	13.67	8.25	7.355
Engines in operation	Main Turbines	Main Turbines	Main Turb. & Cruising Eng.	Main Turb. & Cruising Eng.
Number of Boilers used	4	4	2	2
Burners used (11 per Boiler)(Average)	41.5	21 ⅜	3	3 ⅜
Heating Surface used, Sq. Ft.	21,600	21,600	10,800	10,800
Sq. Ft. per S.H.P.	1.2521	2.9095	6.805	13.697
Cooling Surface, Sq. Ft. (Main Cond.) per S.H.P.	0.6957	1.6166	7.5626	17.445
Pressures (Average):	Stb'd. | Port	Stb'd. | Port	Stb'd. | Port	Stb'd. | Port
Main Steam, at Boilers, pounds gauge	249	248.1	245	242.2
Eng. Room, " "	246.8 | 249.1	245.4 | 247.0	244.4 | 246.6	245.6 | 246.4
Full Speed Ahead Chest, " absolute	215.0 | 215.0	64.4 | 67.9	21.6 | 15.0	14.1 | 13.6
Medium " " " " "	192.4 | 202.6	212.5 | 213.6	36.6 | 29.4	23.3 | 23.5
10th Stage, pounds absolute	31.7 | 32.1	17.1 | 17.0	3.9 | 3.8	1.0 | 1.0
Gland Steam, " gauge	6.8 | 9.8	10.0 | 7.6	3.6 | 3.0	2.4 | 3.0
H.P. Valve Chest, Recp. Eng., pounds gauge	— | —	— | —	223.6 | 214.4	130.0 | 131.1
L.P. Exhaust, " " " absolute	— | —	— | —	41.3 | 40.3	26.6 | 31.1
Vacuum, Main Cond., inches of mercury	28.9 | 29.03	29.00 | 29.30	28.84 | 29.6	29.0 | 29.8
Barometer, inches of mercury	30.2	30.02	30.04	30.05
Auxiliary Steam, pounds gauge	241.3	243.9	241.0	244.3
Exhaust, " "	6.0	7.9	7.8	3.2
Main Feed, at Pumps, " "	300.0	304.0	300.0	330.6
Forced Lubrication, " "	40.9	41.9	60.0	53.0
Fuel Oil, pounds gauge	211.1	207.5	180.0	214.4
Air Pressure in Fire Rooms, inches of water	7.27	4.23	2.15	1.3
Revolutions or Double Strokes per Min. (Average)				
Starb'd Shaft	596.96	418.25	257.25	199.65
Port. "	597.17	419.01	259.37	200.93
Mean	597.06	418.63	258.31	200.29
Main Air Pumps	18.5 | 16.6	21.3 | 22.3	— | 16.5	— | 17.0

REVOLUTIONS OR DOUBLE STROKES PER MIN. (AVERAGE)				
FEED PUMPS	(2) 28.9	(2) 18.15	(1) 8.8	(1) 5.4
FORCED DRAFT BLOWERS	(4) 1,662.0	(4) 1205.0	(1) 918.1	(1) 676.3
FUEL OIL SERVICE PUMPS	(2) 25.1	(2) 10.75	(1) 4.0	(1) 3.3
SHAFT HORSE POWER (AVERAGE)			TOTAL S.H.P. | I.H.P. CR. ENG.	TOTAL S.H.P. | I.H.P. CR. ENG.
STARB'D SHAFT	8,716	3712	826 | 409	364 | 220
PORT "	8,535		761 | 395	324 | 203
TOTAL	17,251	7,424 *	1587 | 804	688 | 423
TEMPERATURES, DEGREES F. (AVERAGE)				
MAIN FEED TANK	70.6	51.5	50.9	50.4
INJECTION	45.7	42.9	34.5	32.0
OUTBOARD DELIVERY	74.7 | 75.3	58.9 | 58.6	42.8 | 42.5	37.3 | 37.1
OUTSIDE AIR	46.0	32.0	38.0	26.0
ENGINE ROOM	73.9	68.5	71.9	84.1
AUXILIARY ROOM	71.6	70.9	69.2	62.3
FIRE ROOMS	68.35	74.7	73.4	66.4
FEED WATER	177.5	190.7	209.1	205.1
FUEL OIL TO HEATERS	45.2	48.1	52.5	48.9
BURNERS	112.7	147.6	163.2	121.9
SMOKE PIPES	504.4	416.6	316.2	301.3
WATER CONSUMPTION				
POUNDS PER HOUR, MEASURED	291,083.70	143,218.4	37,814.9	19,562.1
RESERVE FEED	536.25	834.5	655.0	296.25
TOTAL EVAPORATED	291,619.95	144,052.9	38,469.9	19,858.35
PER S.H.P.	16.874	19.292	23.828	28.433
EVAP'D PER SQ. FT. OF H.S.	13.5	6.67	3.56	1.84
LB. OF OIL	13.042	13.926	13.516	12.792
KNOT RUN	9,846.04	5,994.46	2,466.97	1,626.67
FUEL OIL CONSUMPTION				
B.T.U. PER POUND	19,158	19,158	19,158	19,158
SP. G. AT 60° F.	0.905	0.905	0.905	0.905
POUNDS PER HOUR	22,318.2	10,284.79	2,437.2	1,529.23
KNOT RUN	753.54	427.98	156.29	125.26
AT 29.5 KTS.	745.0	24 KTS. 426.5	15.5 KTS. 155.0	12 KTS. 124.0
GUARANTEED, AT	705.0	445.0	213.0	170.0
BELOW (B) OR ABOVE (A) GUARANTEE	(A) 40.0	(B) 18.5	(B) 58.0	(B) 46.0
HOUR PER S.H.P.	1.2937	1.3853	1.5357	2.2227
SQ. FT. OF H.S.	1.033	0.476	0.226	0.142

* DOUBLED STARB'D. SHAFT. PORT TORSIONMETER OUT OF ORDER.

The water consumption was also carefully measured on these trials.

Trial (*a*).—The standardization trial was run on the measured-mile course off Delaware Breakwater. It was commenced at 2:00 P. M., February 21, 1914, and temporarily discontinued at 5:35 P. M., on account of darkness, eighteen runs having been made. The trial was resumed at 7:00 A. M. the following day and was completed at 9:00 A. M., twelve additional runs being made, Nos. 19 to 30, inclusive. The conditions of weather and sea were favorable throughout the runs.

The data obtained, at the several speeds, is given in Table I, from which the curves, Plate I, were plotted. Runs Nos. 16, 17, 18 and 22 were not used in plotting the curves, conditions being unsatisfactory.

It was deduced from the official revolution-speed curve that the following mean r.p.m. of the two shafts, are required to obtain the speeds noted:

R.p.m.	Knots.
196.9	12
256.7	15.5
417.7	24
593.3	29.5

It was estimated that the displacement at the middle of the five high runs was 1,049 tons, corresponding to a mean draught of 9 feet 6⅛ inches.

Trial (*c*).—The trial was started at 12:35 P. M., shortly after the conclusion of trial (*a*). The weather and sea conditions were fair. It was run in deep water off Five Fathom Bank Lightship and vicinity and was entirely satisfactory. The data obtained are tabulated in Table II.

Trial (*f*).—At 8:50 A. M., February 23, the vessel began the trial, which was run with the cruising engines and turbines in combination. The weather was good and the trial was successfully completed at 12:50 P. M. The trial was con-

ducted in the lower part of Delaware Bay and outside the Breakwater. The data are given in Table II.

Trial (*b*).—This trial, at highest speed attainable, was commenced at 10:35 A. M. on February 26, no trials being held February 24 and 25 on account of bad weather. Conditions were generally favorable. The course was off the coast in a southerly direction between Delaware and Chesapeake Bays. The contract speed of 29.5 knots was slightly exceeded, but the fuel-oil consumption was in excess of the guarantee, as noted in Table II, which also contains the general data of the trial.

Trial (*d*).—Following trial (*b*) the four-hour fuel-oil and water-consumption trial at 15.5 knots, with cruising engines in use, was run off the entrance of Chesapeake Bay, on the afternoon of the same day. The weather conditions were very good and the fuel-oil consumption was readily bettered. For data see Table II.

Trial (*e*).—This trial began at 7:41 P. M., February 26, immediately following trial (*d*). The trial was run outside the entrance of Chesapeake Bay until daylight of the 27th; the remainder of the trial was run in the Bay. The weather conditions were excellent. Throughout the trial the machinery worked well, with few minor exceptions, and the performance was considered satisfactory. The vessel maintained an average of 258.6 r.p.m., corresponding to a speed of 15.61 knots per hour. No other data was taken.

GAS ANALYSIS.

By W. N. Berkeley, Ph. D., Chemist, U. S. Naval Engineering Experiment Station.

The whole rationale of gas analysis depends on the fact that if you bring together, under suitable conditions of temperature, state of subdivision, and of intimate mixture, a combustible material and a supporter of combustion (the masses of these two bearing to one another a certain quantitative relationship) there is formed, as the sole product of combustion, carbon dioxide (CO_2) in the theoretical case of a combustible substance containing carbon alone, or, in addition to these, water (H_2O) in the usual case of a combustible substance containing carbon and hydrogen.

Also, that in the presence of an excess of the "combustible" you change the composition of the products of combustion by the addition of carbon monoxide (CO), frequently in considerable amount, and possibly by the further addition of certain hydrocarbons, which are apt to be produced by the same conditions that give rise to the formation of carbon monoxide.

Likewise, that in the presence of an excess of the supporter of combustion (air), you get with an excess of air such a deficiency of carbon dioxide as to readily reveal the fact that an unnecessary excess of air is being used.

DIFFERENT OPINIONS AS TO WHICH CONSTITUENT OF FLUE GAS SHOULD BE TAKEN AS THE MOST RELIABLE INDEX TO CONDITIONS OF COMBUSTION.

As to which constituent of flue gas should be regarded as the most reliable index to conditions of combustion, there is a considerable difference of opinion, though there seems to be a decided disposition on the part of most authorities to assign this rôle to carbon dioxide.

Hempel ("Gas Analysis," p. 256), however, holds that the conditions of combustion in a furnace are most accurately indicated by the percentage of oxygen in flue gas; while Randall (Bulletin 373, U. S. Geological Survey, page 11) says that carbon monoxide is "in general a good guide to efficient operation."

As has been said, however, an examination of the statements contained both in standard books on gas analysis and in technical and semi-technical journals shows that a preponderance of opinion attributes the dominating influence to carbon dioxide; or, to speak more accurately, that it is the percentage of carbon dioxide that most clearly indicates the character of combustion.

Although this difference of opinion is unfortunate, it should not be thought that it impairs, to any great extent, the unquestioned and unquestionably great value of flue-gas analysis; for, while it may not be possible at the present time to assign to each constituent just exactly the proportionate influence that it exerts on the character of combustion, or rather to determine just what the percentage of each constituent indicates as to the character of combustion, yet there are certain limiting percentages that are assigned to each constituent, as permissible under satisfactory conditions; and it is because flue gas analysis affords a very reliable means of detecting even a slight departure from these limiting values that it is as important an index to the condition of operation of a furnace as is the indicator diagram to that of an engine.

AN OUTLINE OF THE METHOD OF MAKING A FLUE-GAS ANALYSIS.

The process, in brief, consists in measuring accurately any convenient volume of the gas, preferably 100 cubic centimeters, and the passage of this through a series of absorbents, by which various constituents are successively absorbed, the volume of each being determined by the reduction in the volume.

Such determinations are most readily made by means of some form of the well known Orsat type of gas analysis apparatus, such for instance, as is shown in Fig. 1. The details of the analysis, as given below, therefore, will apply especially to this particular form of apparatus; but a thorough understanding of the manipulation of this instrument will permit an easy adjustment of the instructions to any similar instrument.

HOW TO SECURE A REPRESENTATIVE GAS SAMPLE.

No matter how much care and skill are exercised in the analysis of a sample, unless this represents the average composition of the material sampled (or, at least, its composition at the time of sampling), the determination of its composition will be a worse than useless operation, for it will not only furnish any data of value but may lead to entirely erroneous conclusions. In order to secure truly representative samples of flue gas, it is necessary that portions be drawn from a large number of points in the cross-section of the stack, or "pass," *and that there should be a continuous flow of gas through the sampling tube and all connecting tubes between this and the sampler.* In this way small portions are taken from a very large volume (just as in sampling coal, ore, etc., small portions are taken from a great many points in the pile), and in this way the probability of securing an average sample is increased. Such a sample can be secured by means of the device shown in Fig. 2 (seen clearly on the left side of the figure). In this figure A is a steam-jet aspirating pipe joined by means of a rubber tube to the filter flask B, which in turn is joined by a second tube to the pipe C, through which the gas from the stack, or pass, is being aspirated. The gas sampling bottle D is joined to the pipe C, and as the mercury is displaced by the gas it fills a second sampling bottle not shown.

A useful device for drawing a sample from a number of points in the cross-section of the stack, or pass, consists of a series of profusely perforated tubes, radiating from a small

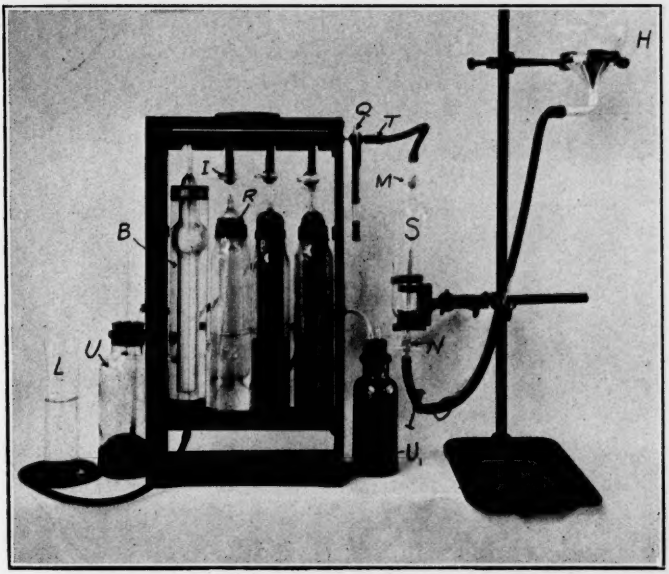


FIG. 1.

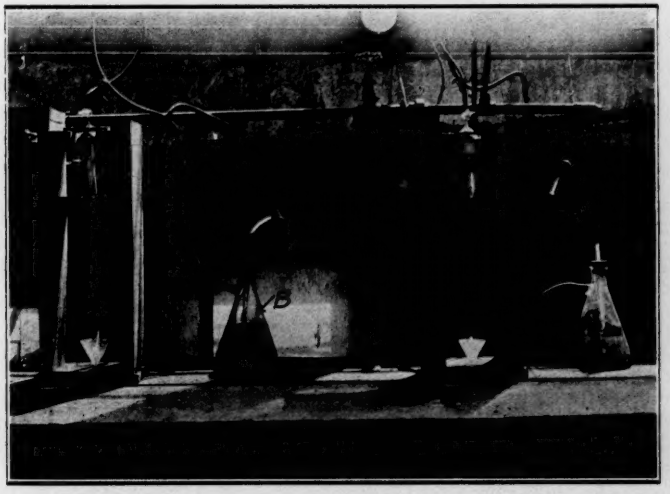


FIG. 2.

A = Steam-jet aspirator pipe.
B = Filter flask.
C = Pipe (gas) leading from stack or "pass."
D = Gas-sampling bottle.

central cylinder, closed at the bottom, but joined at the top to a tube (about ½ inch) which passes through the side of the stack and is joined to the sampling bottle in the manner just described. A very satisfactory form of such sampling bottling is shown in Fig. 1, S.

DRAWING THE SAMPLE FROM THE STACK, OR "PASS."

The sampler having been filled with mercury and attached to the tube through which the gas is being aspirated, the mercury is allowed to flow at a predetermined rate from the lower stop cock, this rate being regulated by means of a capillary tube attached to the stop-cock, so that the time necessary to fill the sampling bottle may be made to vary from a minute or two to several hours. By interrupting the flow of mercury while there is still left a volume of five or ten cubic centimeters, an efficient seal is provided, and the sample may be kept till such a time as it may be convenient to examine it.

PREPARATION OF GAS-ANALYSIS APPARATUS FOR ACTUAL ANALYSIS.

The following directions will be more intelligible if reference is made to Fig. 1. *Filling the absorbing pipettes with the reagents.* The absorbing pipettes consist essentially of two parts: (1) a long glass cylinder constricted at the top, provided with a stop-cock, and filled with small glass tubes which serve both to subdivide the gas and to increase the absorbing surface of the reagent, and (2) a "jacket" containing the reagent into which the cylinder is immersed. To prepare the apparatus for analysis, the rubber band, Fig 1, R, which makes an air-tight connection between the jacket and the cylinder is turned down over the end of the former, which allows of its removal. Two hundred and fifty cubic centimeters of the reagent are placed in the jacket, and the cylinder, with the stop-cock open, is reinserted, the rubber band being replaced so as to make an air-tight seal between the cylinder and its jacket.

After each pipette has been filled in this manner, the next step is to adjust the level of the reagent in each to a point midway between the pipette proper and its stop-cock. To do this, first raise the leveling bottle L till the water in it flows fron T (temporarily disconnected from M), then having closed Q, lower the leveling bottle until the reagent rises to the point indicated. After all the pipettes have been adjusted, the apparatus is ready for the transfer of the sample. Fig. 1 shows the gas analysis apparatus with the sample placed ready for this transfer. After adjusting the reagents, and before connecting with the sampling bottle, the leveling bottle is raised till water flows from T. The apparatus is now ready for the sample.

TRANSFERRING THE SAMPLE TO THE GAS-ANALYSIS APPARATUS.

First, place the leveling bottle L below the level of the apparatus; attach the rubber tube T to M; open the stop-cock and allow the mercury in the capillary of this to fall into the sampling bottle. Now partially open N and allow the mercury to flow from the reservoir of mercury H, thus driving the gas into the measuring burette B. When the water in this has sunk to the 5 c.c. mark raise the leveling bottle till the water in it is level with that in the burette. Reduce the flow of the gas by partially closing N, slowly lowering the leveling bottle so as to maintain the water in it and in the burette at the same level. When the water in the latter reaches the zero mark, close the stop-cock N quickly and then Q. The measuring burette now contains exactly one hundred c.c. of gas, measured at atmospheric pressure.

DETAILS OF THE DETERMINATION OF THE CONSTITUENTS OF THE GAS.

Open stop-cock I so that its bore connects with the *front* capillary of the pipette (next to the measuring burette) which contains a solution of caustic potash (300 grams per litre) and,

lifting the leveling bottle L, pass only enough gas to clear the front capillary of the absorbing reagent; now turn the stop-cock I 180 degrees so as to connect it with the *rear* capillary of the pipette and continue to raise the leveling bottle until the water in B rises to the capillary. Again turn stop-cock I 180 degrees and lower the leveling bottle till the reagent rises to its former level. Repeat this passage of the gas twice (three passages in all). After allowing the water in the measuring burette to drain from its walls, read the level of the water in B. This gives the number of c.c. of carbon dioxide absorbed, therefore the percentage (100 c.c. of the sample having been taken). This procedure is followed in the case of the other pipettes, except that after the gas has been passed through the last pipette (which contains a solution of cuprous chloride in hydrochloric acid) the residual volume should be passed again through the first pipette (caustic potash) to absorb any hydrochloric acid vapors.

The apparatus as shown in Fig. 1 permits of the determination only of carbon dioxide, oxygen, carbon monoxide and nitrogen (by difference). The second pipette (from the left) contains an alkaline solution of pyrogallic acid (for oxygen), and the last one an acid solution of cuprous chloride (for carbon monoxide).

A larger model of this apparatus permits of the additional determinations of hydrogen, of the so-called "heavy hydrocarbons" and of methane. These latter, however, are rarely of any practical importance in flue-gas analysis.

DESCRIPTIONS AND TRIALS OF U. S. TORPEDO-BOAT DESTROYER *NICHOLSON.*

By Lieutenant W. F. Cochrane, U. S. N., Member.

The *Nicholson* is one of six destroyers authorized by an Act of Congress, approved August 22, 1912. These vessels being designated as Torpedo-Boat Destroyers Nos. 51 to 56, inclusive.

The contract for building three of these boats, Nos. 51, 52 and 53, named *O'Brien, Nicholson* and *Winslow,* was awarded to the Wm. Cramp & Sons S. & E. B. Co., Philadelphia, Pa., and was signed December 7, 1912; the time allowed for completion being 23 months for the first boat, 23½ months for the second, and 24 months for the third.

The designed speed was 29 knots, at about 1,050 tons displacement.

The contract price for each vessel was $842,000.00, of which $502,000.00 was allotted for machinery.

HULL.

Principal Dimensions.

Length between perpendiculars, feet and inches	300–0
over all, feet and inches	305–3
Beam, extreme, over plating, feet and inches	30–7
over guards, feet and inches	31–1
molded, feet and inches	30–6
on L.W.L., molded, feet and inches	30–4
Draught, mean, feet and inches	9–5
Displacement, tons	1,050
Area immersed midship section, square feet	196.6
L.W.L. plane, square feet	6,050
wetted surface, square feet	9,760
Displacement per inch at L.W.L.	14.39
Coefficient of fineness, block	0.426
midship section	0.684
L.W.L.	0.660

The general arrangement of the vessel is the same as the *Aylwin* class which were described in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Volume XXVI, No. 1, February, 1914, except that the propellers are located farther forward, eliminating one pair of struts and shortening the shafting. Also the steering engine is located on the main deck in the after house instead of in the pilot house, where it was in the previous vessels.

MACHINERY.

The machinery is similar (except for some details) to that of the *Aylwin* class, which was fully described in the above-mentioned article. The turbines are designed to develop 17,000 shaft horsepower at about 600 revolutions per minute. There are two Cramp-Zoelly turbines arranged on two shafts with a compound, reciprocating, cruising engine on the forward end of each shaft. Between the cruising engine and the turbine there is a hydraulically operated friction clutch. This type of clutch was fully described by its designer, Mr. J. F. Metten, Chief Engineer of the Wm. Cramp & Sons S. & E. B. Co., in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Volume XXVI, No. 1, February, 1914. An improvement in detail over the *Aylwin* type turbine consists in an improved type of nozzle rings. In the *Aylwin* turbine the nozzles were formed of thin steel guide blades in the annular openings in the cast-iron nozzle rings. The guide blades were set in the mold when the nozzle rings were cast. In many cases it was found that the guide-blade angles were not correct, and to remedy this trouble the *Nicholson's* nozzle rings are made of wrought steel with guide blades of the same material as the regular turbine blades. The blades are set in the brass container rings made in sections, and these containers are set in the annular openings of the nozzle rings and are secured by riveting over their edges into recesses provided in the nozzle rings. Conditions encountered in service by the

vessels of the *Aylwin* class have required the following difference among the auxiliaries of the *O'Brien* class vessels:

The installation of a fire and bilge pump in each fireroom; an additional distiller; an air compressor for running pneumatic tools, etc., in addition to the torpedo air compressor. The electric plant was increased from two 10-kw. generators to two 25-kw. generators; the oil-cooler circulating pump was omitted, the circulating water for the oil cooler on the *Nicholson* being taken from the main circulating-pump discharge with an emergency connection from the firemain.

Shafting and Bearings.

Cruising-engine crankshaft, length, feet and inches	5–11 3/16
diameter, inches	5½
diameter of hole, inches	2
Intermediate shaft, length, feet and inches	9– 4½
diameter, inches	5¼
diameter of hole, inches	none.
Rotor shaft, length, feet and inches	18–10 5/16
diameter at bearings, inches	11¼
diameter of hole, inches	7
Line shaft, forward section, length, feet and inches	10– 4½
diameter, inches	8½
diameter of hole, inches	4½
after section, length, feet and inches	20– 2¾
diameter, inches	8½
diameter of hole, inches	4½
Stern-tube shaft, length, feet and inches	25– 7¼
diameter, inches	8⅝
diameter of hole, inches	4½
Propeller shaft, length, feet and inches	22– 1½
diameter, inches	8⅝
diameter of hole, inches	4½
Rotor-shaft bearings, number of	2
diameter, inches	11¼
length, inches	16½
Cruising-engine crankshaft bearings, number of	3
diameter, inches	5½
length, forward, inches	7½
middle, inches	9 3/16
after, inches	11⅞
Intermediate shaft bearings, number of	1
diameter, inches	5¼
length, inches	10

Thrust collars on shaft, number of	9
thickness, inch	¾
space between, inches	1¼
outside diameter, inches	12½
inside diameter, inches	8 13/16
bearing surface, ahead, square inches	493.8
astern, square inches	555.5
Thrust rings, number of, ahead	8
astern	9
effective thrust surface, ahead, square inches	463.0
astern, square inches	520.9
Line-shaft bearings, number of, each side	2
diameter, inches	8½
length, inches	16
Stern-tube bearings, diameter, inches	9½
length, forward, inches	27
after, inches	43
Strut bearings, diameter, inches	9½
length, inches	48 7/16
Main condensers, cooling surface, each, square feet	6,000.9
total, square feet	12,001.8
tubes, number of, each	2,934
outside diameter, inch	⅝
length between tube sheets, ft. and ins.	12–6
thickness, B.W.G.	18
Auxiliary condenser, cooling surface, square feet	306.5
tubes, number of	336
outside diameter, inch	⅝
length between tube sheets, ft. and ins.	5–6¾
thickness, B.W.G.	18
Feed-water heaters, number of	2
heating surface, each, square feet	250
total, square feet	500
tubes, number of, each	238
outside diameter, inch	⅝
length between sheets, ft. and ins.	6–5⅛
thickness, B.W.G.	18
Propellers, twin-screw, machined faces	manganese bronze.
number of blades, each	3
diameter, feet and inches	7–8½
pitch, feet and inches	6–8
pitch ÷ diameter	.865
projected area, square feet	28.21
helicoidal area, square feet	31.50
disc area, square feet	46.67
Boilers, 4 White-Forster	oil burning.
working pressure, pounds per square inch	260

Boilers — external height, feet and inches	12–7 7/16
external length, feet and inches	13–2 13/16
external width, feet and inches	14–7 3/8
number of burners, each	11
heating surface, each, square feet	5,400
total, square feet	21,600
combustion space, each, cubic feet	452
heating surface ÷ combustion space	11.9
tubes, number of, each	3,256
outside diameter, inch	1
thickness, B.W.G.	12
mean exposed length, feet	6.34
Smoke pipes, number of	4
height above oil burners, mean, feet	34
area, each, square feet	16.57
combustion space ÷ area of smoke pipe	27.28
Forced-draft blowers, number of	4
type, Keith fans.	
driven by Terry steam turbines.	
Electric generators, number of	2
rated output, each, kilowatts	25
type, Diehl, driven by Terry steam turbines.	
Evaporators, number of	2
type, Reilly multicoil.	
heating surface, each, square feet	61.50
Distillers, number of	3
type, straight-tube cooling surface, each, sq. ft.	20.54
Fuel-oil heaters, number of	2
type, U-tube.	
heating surface, each, square feet	29.53
Lubricating-oil cooler, type, straight-tube cooling surface, square feet	175.22

Pumps.

Purpose.	Number.	Type.	Diameter, inches.		
			Steam cylinders.	Water or air cylinder.	Stroke, inches.
Main air	2	Blake, twin vertical beam	14	28	18
Main circulating	2	Cramp, engine-driven centrifugal	8	....	6
Main feed	2	Blake, vertical simplex	15	10	16
Auxiliary feed	2	Blake, vertical simplex	15	10	16
Engine-room F. & B	2	Blake, vertical simplex	7	7	12
Fireroom F. & B	2	Blake, vertical simplex	7	7	12
Auxiliary condenser, air and circulating	1	Blake, horizontal simplex, comb	6	8	7
Evaporator feed	1	Blake, vertical simplex	4½	6	6
Distiller fresh water	1	Blake, vertical simplex	3½	4	4
Lubricating oil	2	Blake, vertical simplex	4½	6	6
Fuel-oil service	4	Blake, vertical duplex	6	3½	8
Fuel-oil booster	2	Blake, vertical simplex	4½	6	6
Air compressor	1	Westinghouse, vertical simplex	11	11	12

TRIALS.

The contract required:

(*a*) A progressive trial over the measured-mile course at Lewes, Del., for standardizing the screws, extending from maximum speed (at least 29 knots) down to a speed of 8 knots.

(*b*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed attainable, the average for the four hours not to be less than 29 knots. The speed to be determined by the average revolutions of the main shafts, according to the official standardization curve.

(*c*) A fuel-oil and water-consumption trial of four hours' duration in the open sea in deep water, at an average uniform speed of 24 knots, as nearly as possible. The trial to be conducted as nearly as possible to service cruising conditions.

(*d*) A fuel-oil and water-consumption trial of four hours' duration at 15½ knots, under conditions similar to the preceding trial, but with the cruising engines connected and in use.

(*e*) An endurance trial of ten hours' duration in the open sea at an average uniform speed of 15½ knots, as nearly as possible, following as closely as possible trial (*d*), with cruising engines connected and in use. Fuel oil and water consumption will not be measured on this trial, the purpose of which is to determine the reliability and endurance of the cruising engines.

(*f*) A fuel-oil and water-consumption trial of four hours' duration in the open sea, with the cruising engines connected and in use, at an average uniform speed of 12 knots, as nearly as possible.

(*g*) In addition to the above-enumerated trials the contract was amended to include a two-hours' trial at about 15½ knots, with the main turbines only in use, fuel oil and water consumption to be carefully measured on this trial.

Fuel-Oil Consumption Guarantees.—The contractors guaranteed that the fuel-oil consumption per knot run for all purposes, including that necessary for all auxiliaries in use on the

trials, would not exceed 692.25 pounds at guaranteed maximum speed, 444.6 pounds at 24 knots, 213 pounds at 15½ knots, and 170 pounds at 12 knots, the consumption of the fuel oil at these speeds to be determined by the Trial Board from a curve based on the rate of fuel oil consumed on trials (*b*), (*c*), (*d*) and (*f*), and corrected to a standard of 19,500 B.t.u. per pound of fuel oil.

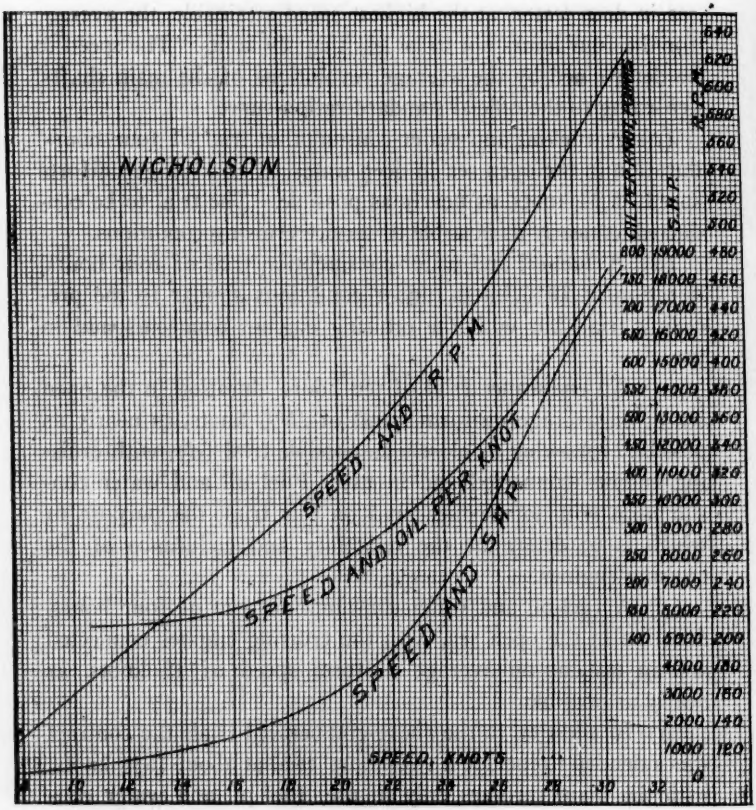


PLATE I.

Standardization Trial (a).—This trial was conducted under favorable weather conditions on the measured mile at Lewes, Delaware. Twenty-eight runs were made over the course at

various speeds, and from the data obtained the curves in Plate I were plotted. Table I gives the standardization data.

From the curve in Plate I the following r.p.m of the propellers were found to be necessary for the various speeds:

12 knots	195.00
15½ knots	252.00
24 knots	415.40
29 knots	563.40

Four-Hour Full-Speed Trial (b).—This trial began at 8:40 A. M., March 24, 1915, off Five Fathom Bank Lightship, and was completed at 12:40 the same day.

The weather was fair. The data obtained on this run is given in Table II. The machinery operated excellently. All guarantees were easily attained.

Four-Hour 24-Knot Fuel-Oil and Water-Consumption Trial (c).—Following the full-speed trial, the 24-knot trial began at 1:35 P. M., and was completed at 5:35 P. M. The weather was fair. The trial was very successful. For data, see Table II.

Two-Hour 15½-Knot Fuel-Oil and Water-Consumption Trial (g).—The trial began at 5:50 P. M., following the 24-knot run, and was completed at 7:50 P. M. The trial was very successful. Data is given in Table II.

Four-Hour 12-Knot Fuel-Oil and Water-Consumption Trial (f).—This trial commenced at 6:10 A. M., March 25, 1915, and was completed at 10:10 A. M. the same day. The weather was fair and the trial very successful. For data, see Table II.

Four-Hour 15½-Knot Fuel-Oil and Water-Consumption Trial (d).—The trial began at 10:15 A. M., March 25, 1915, and was completed at 2:15 P. M. the same day. The trial was very successful. The weather was fair. For data, see Table II.

Ten-Hour 15½-Knot Endurance Trial (e).—This trial began at 2:30 P. M., March 25, 1915, and was completed at 12:30 A. M., March 26, 1915. The weather was fair. The trial was very successful, and the reciprocating engines and clutches operated excellently.

TABLE I.—STANDARDIZATION U. S. S. "NICHOLSON," LEWES, DELAWARE, MARCH 23, 1915.

No. of Run.	Speed.	R.P.M.			S.H.P.			Speed, Average group.	S.H.P., Average group.	R.P.M. Average group.
	Knots.	*Stbd.*	*Port.*	*Mean.*	*Stbd.*	*Port.*	*Mean.*			
1	7.27	140.63	140.48	140.56	141	142	283			
2	9.82	140.43	140.96	140.70	147	150	297	8.69	299	141.44
3	7.83	143.65	143.91	143.78	165	153	318			
4	13.00	199.74	199.91	199.83	379	385	764			
5	11.64	198.72	198.30	198.51	397	371	768	12.25	758	199.09
6	12.71	199.67	199.30	199.49	369	363	732			
7	15.33	254.17	253.31	253.74	813	756	1,569			
8	15.70	252.90	253.02	252.96	758	743	1,501	15.58	1,542	253.45
9	15.58	254.01	254.23	254.12	800	798	1,598			
10	20.62	342.20	340.64	341.42	1,915	1,897	3,812			
11	20.91	340.94	341.29	341.12	1,857	1,866	3,723	20.73	3,738	341.30
12	20.48	341.97	341.06	341.52	1,863	1,830	3,693			
13	24.78	426.52	423.48	425.00	3,985	3,837	7,822			
14	23.78	420.85	419.31	420.08	3,827	3,735	7,562	24.25	7,632	421.56
15	24.66	421.62	420.55	421.09	3,813	3,768	7,581			
16	25.62	467.64	468.06	467.85	5,328	5,259	10,587			
17	26.51	468.56	471.90	470.23	5,291	5,350	10,641	26.10	10,604	469.41
18	25.75	469.20	469.37	469.29	5,345	5,203	10,548			
19	29.36	566.65	553.48	560.07	8,211	7,648	15,859			
20	28.66	569.75	559.88	564.82	8,484	7,765	16,249	28.98	16,049	562.77
21	29.22	564.01	558.74	561.38	8,088	7,749	15,837			
22	30.00	607.17	605.32	606.25	9,314	8,671	17,985			
23	30.74	610.83	606.33	608.58	9,553	8,869	18,422			
24	30.15	610.48	611.03	610.76	9,517	8,567	18,084	30.51	18,313	611.22
25	30.85	615.79	610.91	613.35	9,569	8,874	18,443			
26	30.53	619.14	617.15	618.15	9,714	8,903	18,617			
27	14.49	.....Backing.								
28	14.12									

TABLE II.—U. S. S. "NICHOLSON."

	4 hour full power.	4 hour 24 knot.	4 hour 15½ knt.	4 hour 12 knot.	2 hour 15½ knt.
Date of trial	3/24/15	3/24/15	3/25/15	3/25/15	3/24/15
Displacement	1,045	1,047.5	1,044.5	1,051.5	1,046.9
Boilers in use	4	4	2	2	2
Heating surface	21,600	21,600	10,800	10,800	10,800
Speed, in knots	29.084	24.081	15.625	12.13	15.621
R.p.m., starboard	566.40	417.35	253.91	197.15	253.85
port	565.79	417.06	253.80	196.98	253.97
mean	...	417.21	253.88	197.07	253.91
S.H.P., mean	566.10	7,257	1,624	866	1,503
I.H.P., cruising engines	15,906	...	723	375	...
Pressures:					
Main steam (G)	250.8	248.2	251.9	250.6	222.8
Main turbines:					
Full speed (abs.) S.	201.3	70.4	17.4	10.0	22.0
Full speed (abs.) P.	215.4	75.0	15.0	10.0	23.2
Cruising (abs.) S.	149.9	195.5	37.0	18.0	67.6
Cruising (abs.) P.	164.8	209.0	38.4	19.4	72.9
10th stage (abs.) S.	24.2	12.6	3.0	1.1	4.8
10th stage (abs.) P.	26.2	12.6	3.3	1.6	4.8
14th stage (abs.) S.	14.6	7.3	.5	.56	2.4
14th stage (abs.) P.	14.0	6.6	1.5	.50	2.5
Gland steam gage	6.3	4.9	3.0	2.1	4.7
Cruising engines:					
H.P. valve chest (G)	...	...	221.5	223	...
			224.6	137.8	...
L.P. receiver (G)	...	...	28.8	7.4	...
			30.0	9.3	...
Vacuum, starboard	26.6	28.3	28.3	28.5	27.6
port	27.8	28.8	29.0	29.0	28.2
Auxiliary exhaust	8.0	5.1	4.0	5.6	5.3
Forced lubrication system	12.6	8.4	10.4	9.9	10.0
Oil to clutches	...	...	75.0	75.0	...
Temperatures:					
Air	49.5	47.4	48.6	44.1	46.4
Engine room	80.0	71.3	86.3	79.1	70.1
Auxiliary room	66.8	69.5	63.3	62.5	65.3
Fireroom	71.7	71.7	85.0	84.7	77.1
Oil to cooler	104.9	95.6	70.1	64.0	85.0
Oil from cooler	84.9	75.9	41.6	41.0	66.0
Main injection	46.0	42.0	49.0	39.0	40.0
Main discharge	76.1	62.6	50.0	49.0	54.5
Fuel oil to heaters	47.0	62.5	70.0	...	...
Fuel oil from heaters	57.1	95.6	...	69.0	66.6
Smoke pipes	...	...	...	...	...
R.p.m. blowers	1,495	110.0	750.6	580	...
Air pressure	6.2	2.5	1.3	1.0	1.9
Water consumption, pounds per S.H.P., M.E.	16.224	18.632	22.06	24.22	33.94
Fuel oil, per knot run	687.3	397.74	155.95	131.86	222.42
Water evap. per pound oil	12.9	14.12	14.7	13.12	14.69
Propeller data, diameter, ins.	92½	...	...	...	...
pitch, inches	80	...	...	...	...

THE PNEUMERCATOR.

A DESCRIPTION OF ITS INSTALLATION AND OPERATION.

By Henderson B. Gregory, Associate.

GENERAL DESCRIPTION.

The Pneumercator was invented by Mr. Harry S. Parks, of Philadelphia, Pa., with intention of providing an apparatus that would measure the depth and volume of the contents of tanks, reservoirs, etc., with extreme accuracy and register this measurement at any convenient place at any distance from such tank or reservoir.

It is now manufactured in various models to meet the requirements of its application, all of which work on the same principle and consist of essentially the same elements.

These elements are—

(1) A balance chamber;

(2) A mercury gage, which is calibrated in feet and inches and the corresponding weight or volume;

(3) A pump or other means of furnishing compressed air;

(4) A control valve, directly attached to the mercury gage and also connected, through small piping, to the balance chamber and air pump.

Figure 1 shows a diagram arrangement of the component parts, and Figure 2 a detail of the control valve.

By placing the handle of the control valve in its various positions, connection is made exclusively, (*a*) between the balance chamber and the air pump, or (*b*) between the balance chamber and the gage, or (*c*) the reading of the mercury gage can be returned to zero without losing the pressure

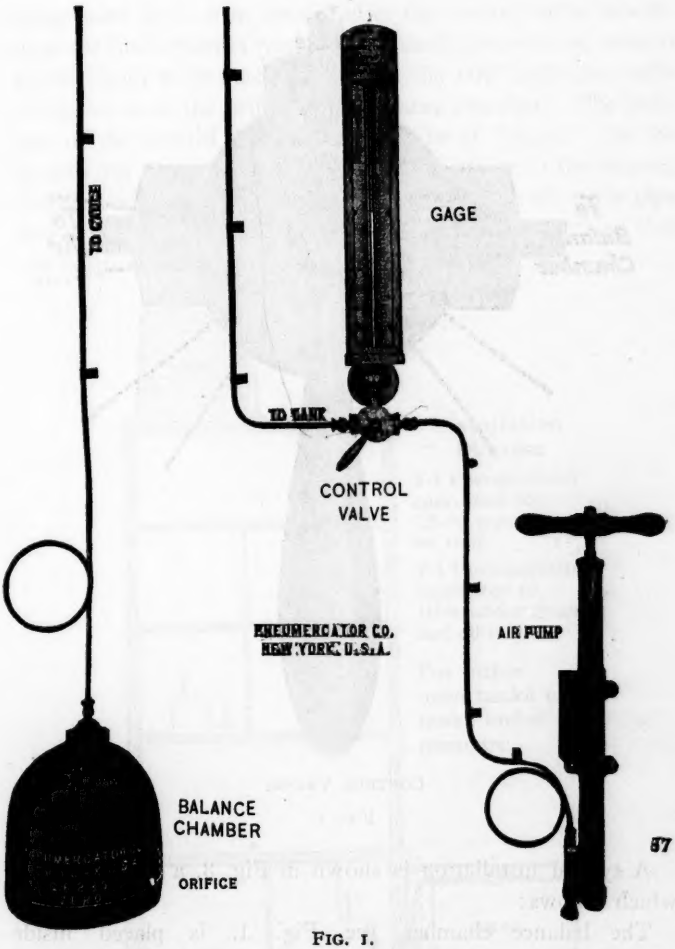
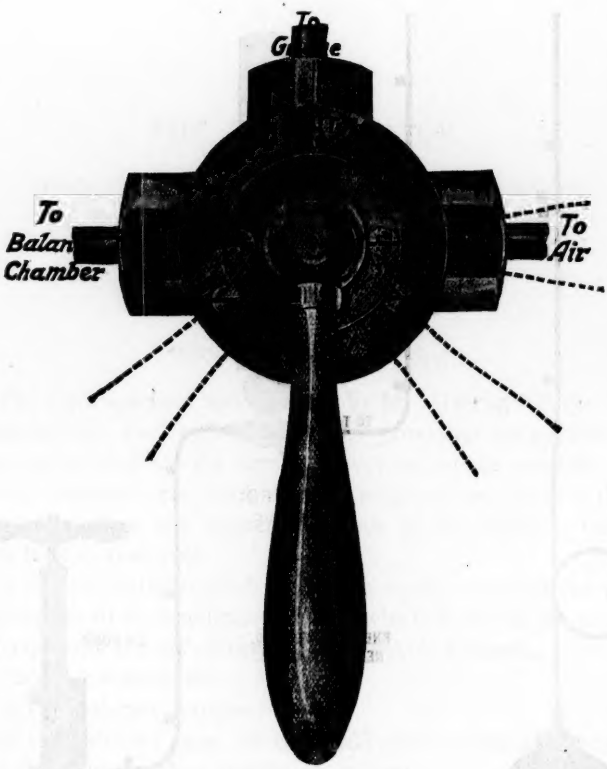


FIG. 1.

then existing between the balance chamber and the mercury gage. There is a fourth position (*d*) in which the various connections are all closed. The various positions of the valve handle, for these several functions, are defined by an index on the valve marked "GAGE"—"SHUT"—"AIR"—"VENT," corresponding to (*b*), (*d*), (*a*) and (*c*), respectively.



CONTROL VALVE.

FIG. 2.

A typical installation is shown in Fig. 3, a description of which follows:

The balance chamber, see Fig. 1, is placed inside the tank and securely attached so that the orifice is at some predetermined point, which point is desired to be the zero or lowest reading on the scale of the mercury gage. The pipe line is then led to the place where the reading of the contents of the tank is desired to be taken and connected to the control valve of the mercury gage. The source of

compressed air is then connected to the control valve, whereupon the instrument is ready to be placed in operation, as soon as the liquid to be measured enters the tank and rises sufficiently to cover the orifice in the balance chamber. The position of the control valve should then be at "gage." As the liquid rises in the tank it flows into the orifice in the balance chamber until its weight or head compresses the air in the pipe line (which is now sealed at the top by the mercury) so that one balances the other.

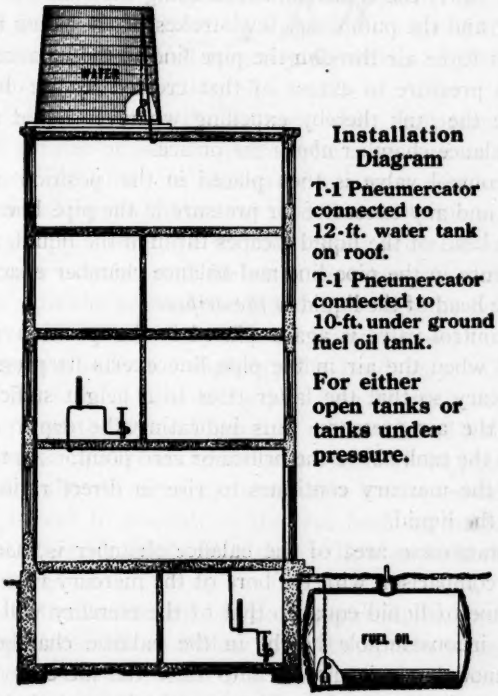

WATER
FUEL OIL
Installation Diagram
T-1 Pneumercator connected to 12-ft. water tank on roof.
T-1 Pneumercator connected to 10-ft. under ground fuel oil tank.
For either open tanks or tanks under pressure.

FIG. 3.

The point at which this balance takes place varies with the length of the connecting pipe line, and consequently the accuracy of the reading of the mercury gage cannot be depended

upon unless the point of balance takes place *exactly* at the orifice, as the mercury gage will show an error proportionate to the distance at which the liquid ***in the balance chamber*** stands above the orifice.

It will be readily seen, however, that if the level in the balance chamber where the air and liquid meet is established at the orifice, the mercury in the gage will indicate the true depth of the liquid in the tank above the orifice. Therefore, to accurately set the gage, the control valve is placed in the position marked "air," the connection then being between the balance chamber and the pump. A few strokes of the pump are sufficient to force air through the pipe line to the balance chamber at a pressure in excess of that created by the head of liquid in the tank thereby expelling whatever liquid may be in the balance chamber *above* the orifice.

The control valve is then placed in the position marked "shut," and any excess of air pressure in the pipe line greater than the head of the liquid escapes through the liquid, and the air pressure in the pipe line and balance chamber exactly balances the head of the liquid *at the orifice.*

The control valve is again placed in the position marked "gage," when the air in the pipe line exerts its pressure on the mercury so that the latter rises to a height sufficient to balance the air pressure, thus indicating the depth of the liquid in the tank above the orifice or zero point. As the tank is filled the mercury continues to rise in direct ratio to the head of the liquid.

The transverse area of the balance chamber is made very large in comparison with the bore of the mercury tube so that the volume of liquid equal to that of the mercury will occupy only an inconsiderable height in the balance chamber. Of course enough air is displaced to raise the mercury column and an equal volume of liquid enters the balance chamber to take its place, but this volume is comparatively small, and when spread out to the area of the balance chamber is of a depth so slight as to be incapable of measurement, so that no inaccuracy results therefrom.

Such displacement of air through the tubing is so slight and the motion of the mercury rising to the height which balances the pneumatic pressure is so soon accomplished that almost instantly after connection is made between the balance chamber and the indicator, the pressure becomes *static*. Thus the friction of the air in the tubes and chambers has no vitiating effect and the indicator may be located at any distance from the source of pressure.

As the indication is effected through pneumatic pressure of a gas of which the weight is negligible, the indicator may be located below the body of liquid as well as above it and will show an equal degree of accuracy regardless of the length or direction of the connecting pipe line.

The apparatus may also be used when the pressure in the tank is greater or less than atmospheric as well as under atmospheric pressure. In such cases the upper part of the tank is connected by an additional pipe line with the top of the mercury tube. This pipe line carries the pressure or vacuum as the case may be; and the usual pipe line, the pressure or vacuum plus the head of the liquid, and both the liquid to be measured and the mercury column being exposed to the same pressure, the indicator registers the head *only*. Of course should there be a leak in the pipe line between the balance chamber and the control valve or in the control valve itself, the pneumatic pressure caused by the head of the liquid would escape and the mercury gage would give a false reading, therefore it is well to re-establish the true head by means of the pump or other air supply from time to time as the tank is being filled or emptied.

In case the instrument has not been read for some time it is always advisable to "blow out" the system to see if any leak has occurred; if after this is done and the control valve returned to "gage" position the reading is the same as before, no further test is necessary, but if the second reading is less than the first, the control valve should be placed in the "vent" position; this allows the pressure on top of the mercury in the mercury cistern to escape and the mercury (then being under

atmospheric pressure) returns to zero. The control valve is then turned to "gage," and if the mercury rises to the same height as it did first, after "blowing out," these readings are the correct ones.

By these two means the instrument in itself provides a positive check of its accuracy.

INSTALLATION ON U. S. S. "NEW YORK."

The Pneumercator was first brought to the attention of the Navy Department early in the year 1913, and, following a test of its accuracy conducted at the Navy Yard, New York, the results of which aroused considerable interest and speculation, an installation was authorized on the battleship *New York,* for a practical trial aboard ship under service conditions, in connection with the fuel-oil tanks.

The type of instrument installed is shown in Fig. 4, and Fig. 5 illustrates, in diagram, the general method of connecting the apparatus to the inner-bottom tanks aboard ship.

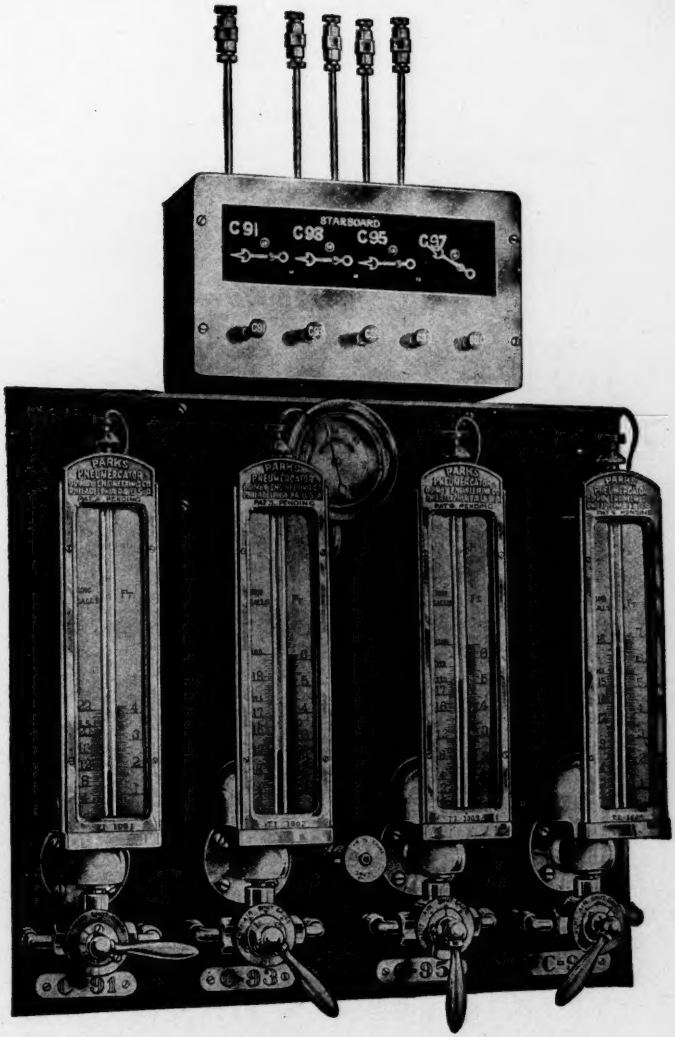
The fuel oil on the *New York,* which is intended only for auxiliary service, is carried in eight double-bottom tanks of various sizes and depths.

A balance chamber was installed in each tank, from which a ¼-inch outside diameter lead pipe was led through an oil-tight connection in the tank tops, from which point a copper pipe of the same size was carried to the engineers' office on the gun deck of the vessel.

The indicating gages were mounted on slate boards attached to the athwart-ship bulkhead.

A connection was made from the ship's compressed-air line, through a reducing valve, to two needle valves mounted on the boards, which in turn admitted the air to the control valve of each instrument and a pressure gage connected with the air line to register the air pressure available.

The scales of the mercury gage were calibrated on one side in feet, inches and half inches, and on the other in the corresponding gallons of oil at 60 degrees F., temperature. At



ONE OF THE SETS OF INSTRUMENTS ORIGINALLY INSTALLED ON U. S. S. "NEW YORK." (NOW REPLACED, EXCEPT ANNUNCIATOR, BY LATEST MODEL, FIG. 7.)

FIG. 4.

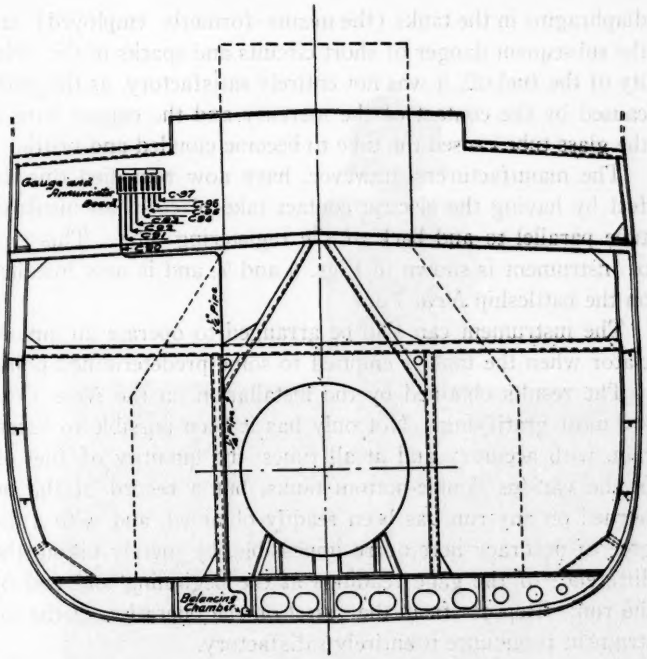


DIAGRAM ARRANGEMENT OF METHOD OF CONNECTING THE APPARATUS TO THE INNER-BOTTOM TANKS ABOARD SHIP.

FIG. 5.

the suggestion of the manufacturer an annunciator was mounted on each board which showed a signal and rang a bell when each tank was filled to the 95 per cent. full point.

To accomplish this a low voltage electric current was connected to the mercury cistern of each instrument, and a copper wire was inserted into the mercury tubes from the top reaching down to a point corresponding to the 95 per cent. full mark on the scales; consequently when the mercury rose to this point the circuit was closed and the annunciator operated.

While this method of ascertaining when the oil had reached a predetermined height was considered of the greatest value, as it did away with the necessity of all electric wiring to the tanks themselves and electrical connections to floats or

diaphragms in the tanks (the means formerly employed) and the subsequent danger of short circuits and sparks in the vicinity of the fuel oil, it was not entirely satisfactory, as the spark caused by the contact of the mercury and the copper wire in the glass tube caused the tube to become clouded and brittle.

The manufacturers, however, have now remedied this defect by having the electric contact take place in an insulated tube parallel to and back of the registering tube. This type of instrument is shown in Figs. 6 and 7, and is now installed on the battleship *New York*.

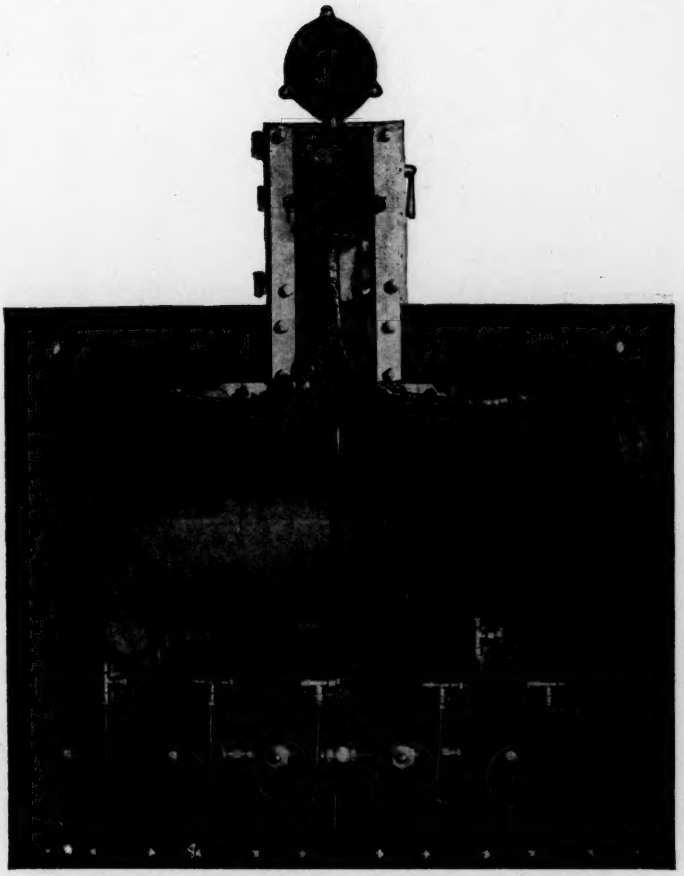
The instrument can also be arranged to operate an annunciator when the tank is emptied to some predetermined point.

The results obtained by the installation on the *New York* are most gratifying. Not only has it been possible to ascertain, with accuracy and at all times, the quantity of fuel oil in the various double-bottom tanks, but a record of the oil burned on any run has been readily obtained, and with a degree of accuracy heretofore impossible, by merely taking the difference of the gage readings at the beginning and end of the run. Reports from the vessel on the operation of the instrument pronounce it entirely satisfactory.

DRAUGHT INDICATOR FOR VESSELS.

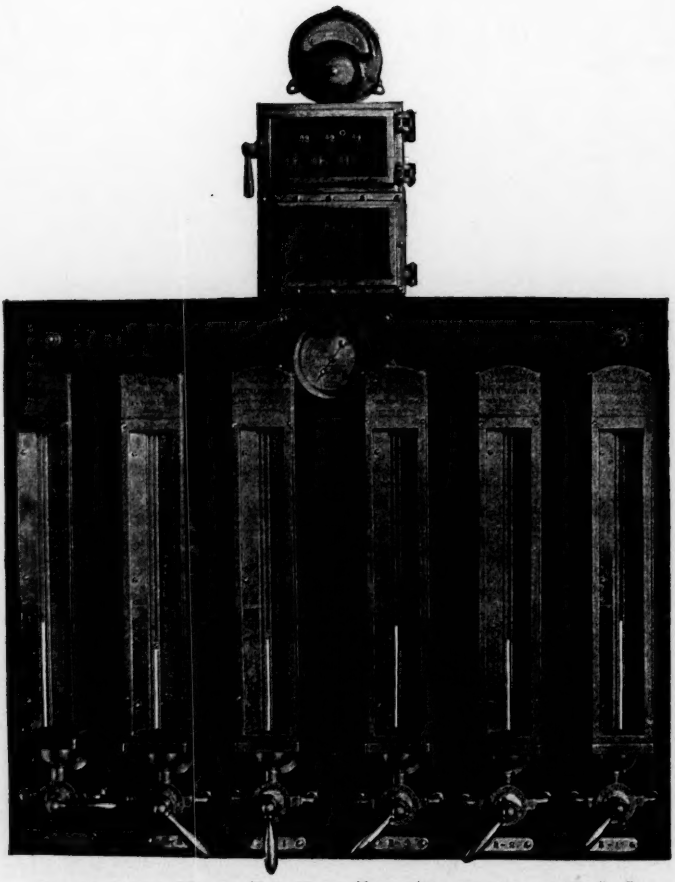
Another application of the pneumercator is that of measuring the draught of vessels. This instrument has not as yet been tested by the Navy, but the reports received from the owners of various merchant ships in which it has been installed are so interesting that a brief description of the instrument and its operation seem desirable.

In this type of instrument, shown in Fig. 8, the balance chamber is placed in a casing, which in turn is connected with the sea itself by means of a 1-inch sea valve. Two balance chambers are used, one forward and the other aft, the sea connection being installed below the light draught trim of the vessel. The ¼-inch copper tubing is then led from the balance chambers to the captain's office or the chart room and



REAR VIEW, MODEL T-I, NAVY TYPE, SHOWING ELECTRIC WIRING AT TOP NECESSARY FOR OPERATING HIGH OR LOW ALARM.

FIG. 6.



FRONT VIEW, MODEL T-I, NAVY TYPE (INSTALLED ON U. S. S. "OKLAHOMA," "PENNSYLVANIA," "ARIZONA" AND E. U. A. DREADNAUGHT "RIVADAVIA.")

FIG. 7.

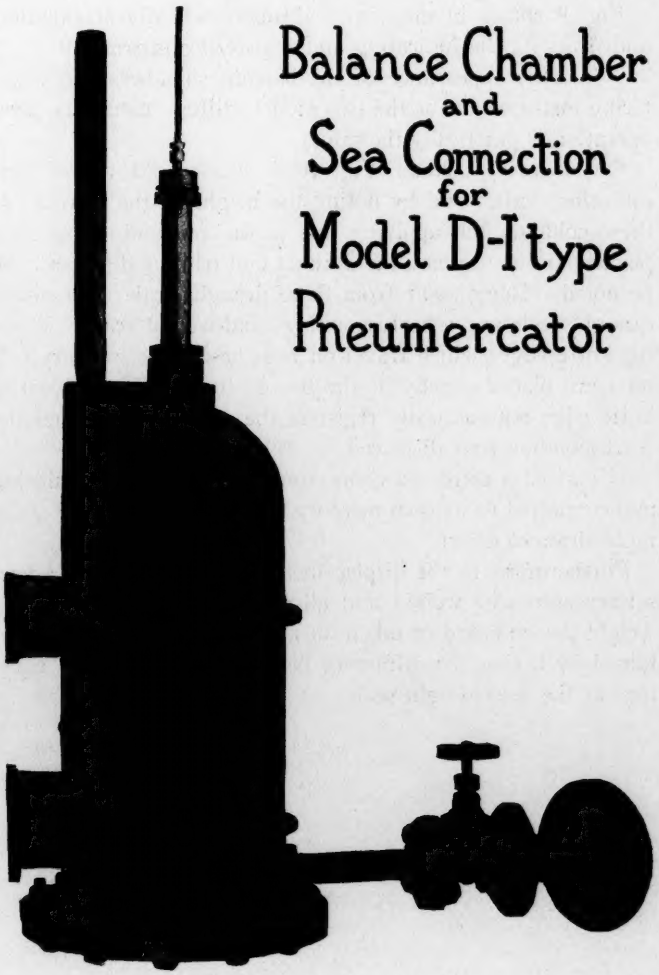


FIG. 8.

connected to the registering portion of the instrument in the same manner as in the T-I type of instrument, Fig. 7, previously described.

Fig. 9 shows in diagram a draught-indicator installation, and, Fig. 10, the indicating and registering instrument.

While the appearance of the balance chambers and registering instruments for the two models differs materially, their operation is practically the same.

One mercury column, Fig. 10, is marked "forward" and the other "aft," and by noting the height of the mercury in these columns and applying this to the feet-and-inches scale placed parallel to them, the draught and trim of the vessel can be noted. Equidistant from these draught scales is a mean-draught scale and a corresponding dead-weight scale. When the knife edges which travel on rods beside the mercury column are placed exactly at the top of the mercury, a central knife edge automatically registers the mean draught and the corresponding tons displaced.

If desired a third sea connection can be installed amidship and connected to its own mercury column, which will give the mean draught direct.

Furthermore, as the displacement or dead weight of a vessel represents its weight and all it contains, the amount of weight put on board or taken off a vessel can be readily ascertained by taking the difference between two successive readings of the dead-weight scale.

Diagram showing Pneumer
Installed on S.S. Herma

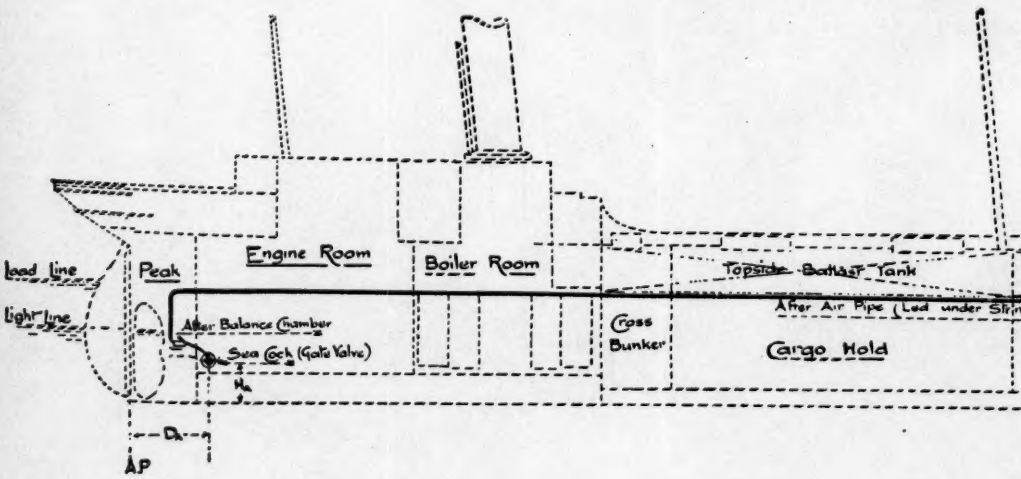
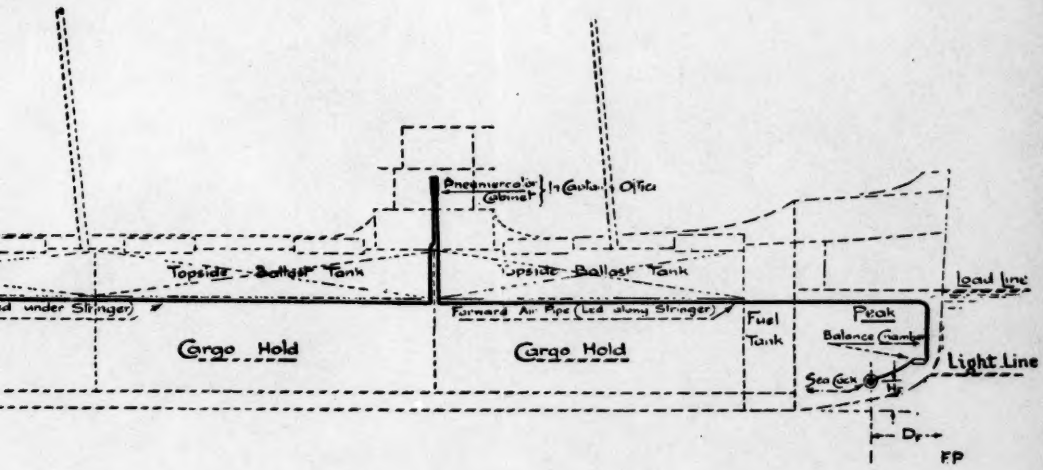


DIAGRAM OF DRAUGHT INDICA

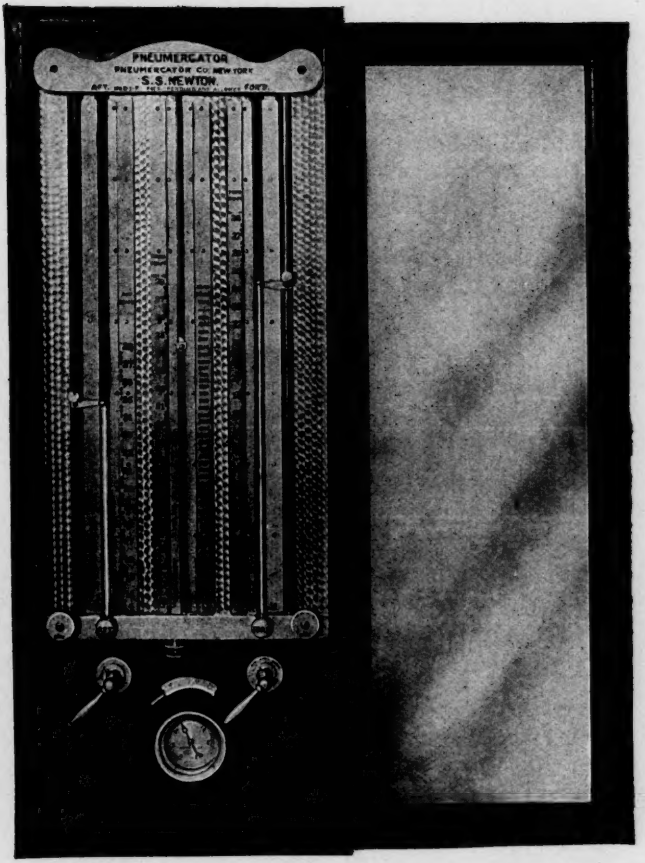
FIG. 9.

umercator Model D-1
lerman Frasch



GHT INDICATOR INSTALLATION.

FIG. 9.



REGISTERING INSTRUMENT FOR DRAUGHT INDICATOR.

FIG. 10.

DESCRIPTION AND TRIALS OF THE TORPEDO-BOAT DESTROYER *DOWNES*.

By LIEUTENANT (J. G.) R. M. GRIFFIN, U. S. NAVY, MEMBER.

The *Downes*, destroyer No. 45, was authorized by Act of Congress approved March 4, 1911. The other ships built under this appropriation are the destroyers *Balch*, *Benham*, *Parker*, *Aylwin*, *Cassin*, *Cummings* and *Duncan*.

The *Downes* is a twin-screw vessel fitted with a combination of Curtis turbines and reciprocating engines, and designed for a speed of 29 knots at a displacement of 1,072 tons, with main engines alone developing 16,000 shaft horsepower. She was built under contract by the New York Shipbuilding Company, Camden, N. J. The contract was signed September 8, 1911, the price being $777,500 and the time of delivery twenty-four months.

The keel was laid June 27, 1912, and on November 8, 1913, the *Downes* was launched. The preliminary official trials were held from January 5 to January 9, 1915, at Lewes, Del., and the vessel was delivered to the Government on February 11, 1915.

HULL.

The arrangement of quarters, armament and equipment is practically the same as that of the other boats of this class, which were described in the February, 1914, number of the JOURNAL in an article on the U. S. S. *Aylwin*, *Parker* and *Benham*. The following summary gives the principal hull data:

Length over all, feet and inches	305–03
between perpendiculars, feet and inches	300–00
Breadth, molded on 9 ft. 3 ins. N.W.L., feet and inches	30–09
extreme over guards, feet and inches	31–01
Displacement for L.W.L., in tons	1,126
Tons per inch L.W.L.	14.6
Load draught, feet and inches	9–11

Wetted surface, load, in square feet	9,760
Area of midship section, in square feet	189.3
Midship section coefficient	0.671
Block coefficient	0.418
Prismatic coefficient	0.622
Capacity of fuel-oil tanks, in tons	307.64
reserve feed-water compartments, tons	36.05

PROPELLING MACHINERY.

The propelling machinery consists of a combination of turbines and reciprocating engines. There are two shafts, each shaft having one Curtis turbine and a compound reciprocating engine. The reciprocating engines are designed for cruising below 15½ knots, and are coupled to the turbines by a mechanical jaw clutch, which can be thrown out of gear for higher speeds.

The cruising engines are fitted with Stephenson links, are reversible, and there is a shift valve worked by the rock shaft for shifting the exhaust from the ahead turbine to the astern turbine.

MAIN TURBINES.

The turbines are of the Curtis marine type. Each turbine was designed to develop 8,000 S.H.P. at 550 r.p.m., corresponding to a speed of 29 knots. The astern turbines are carried in the after end of the same casing as the ahead turbines and are designed to develop 50 per cent. of the ahead power.

Turbine Data.

Ahead turbine :	
1st stage nozzles :	
16 regular, throat area, square inches	10.52
3 cruising, throat area, square inches	1.2
2d stage nozzles :	
81 openings, throat area, square inches	24.22
3d stage nozzles :	
85 openings, throat area, square inches	33.74
4th stage nozzles :	
111 openings, throat area, square inches	38.183
Tip clearance buckets and casings, 5th to 36th stages, inch	0.$\frac{8}{32}$
Tip clearance nozzles and drums, 5th to 36th stages, inch	0.08
Clearance, forward side moving blades, 1st, 2d, 3d stages, inch	0.02
Clearance, forward side moving blades, remaining, inch	0.$\frac{1}{16}$
Clearance, after side moving blades, inch	0.16 to 0.28

Astern turbine:

1st stage nozzles:

10 openings, throat area, square inches	6.75
Tip clearance, buckets and casings, 2d to 10th, inch	0.$\frac{3}{32}$
Tip clearance, nozzles and drum, 2d to 10th, inch	0.08
Clearance, forward side of moving blades, inch	0.06
Clearance, after side of moving blades, inch	0.28

CRUISING ENGINES.

There are two vertical, inverted, direct-acting, compound reciprocating engines, in enclosed casings. They are fitted with Lovekin improved assistant cylinders, suspension links and reversing gear, and a shift valve to change the exhaust from the ahead to the astern turbine. The links and shift valve may be worked by hand or by steam.

Stroke, inches	14
Length of connecting rod, inches	35
Diameter of piston rods, inches	3
Cylinder, diameter, inches	H.P., 12$\frac{1}{2}$; L.P., 26$\frac{1}{2}$

	H.P.		L.P.	
	Top.	Bottom.	Top.	Bottom.
Piston clearance, inch	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{8}$	$\frac{3}{16}$
Travel of valve, inches	4$\frac{1}{2}$		4$\frac{1}{2}$	
Number and size of valves inches	One, 8$\frac{1}{2}$		One, 11	
Kind of valve	Piston, single port.		Piston, double port.	
Valve takes steam	Inside.		Outside.	
Depth of ports in liners, inches	1$\frac{1}{8}$	1$\frac{1}{8}$	1$\frac{7}{64}$	1$\frac{7}{64}$
Steam lap, inches	1$\frac{5}{32}$	$\frac{61}{64}$	1$\frac{9}{64}$	1
Exhaust lap, inch	$-\frac{3}{16}$	$+\frac{1}{4}$	$-\frac{13}{32}$	$+\frac{1}{32}$
Angular advance	38°-40′	38°-40′	44°-15′	
Steam lead, angular	8°	13°-15′	13°-30′	17°-30′
linear, inch	$\frac{1}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{9}{16}$
Cut off, in inches	10.08	9.52	9.52	8.68
p.c. of stroke	72	68	68	62
Mean cut off p.c. of stroke	70		65	
Comp'sion, in inches	1.4	1.68	1.4	1.68
p.c. of stroke	10	12	10	12
Release, in inches	1.59	1.28	2.45	2.24
p.c. of stroke	11$\frac{3}{8}$	9$\frac{1}{2}$	17$\frac{1}{2}$	16
Steam opening, inches	1$\frac{3}{32}$	1$\frac{19}{64}$	($1\frac{7}{64}+1\frac{7}{64}$)	
Exhaust opening, inches	1$\frac{1}{8}$	1$\frac{1}{8}$	2$\frac{7}{32}$	2$\frac{7}{32}$
Steam velocity, liner	4,070	3,260	5,410	5,340
cyl. port	3,830	3,610	6,290	6,210
Exhaust velocity through liner	3,280	3,120	5,390	5,340
R.P.M. (designed)	254			

SHAFTING AND BEARINGS.

The following table gives the data for one shaft with the sections in their proper order from forward:

Name of Section.	Shafting.			Bearings.			
	Length.	Diameter.	Axial Hole.	Number.	Diameter.	Length.	Type.
	ft. & ins.	*ins.*	*ins.*		*ins.*	*ft. & ins.*	
Cruising engine, crank shaft	8–11½	08	3¾	4	08	0–09½	Babbitted
Clutch coupling	18½	08	4	None			
Turbine shaft	22–00¾	16½	9	2	13	0–18	Babbitted
Line shaft, for'd section	11–04	10	7	None			
Line shaft, after section	23–06½	10	7	2	10⅜	0–14	Babbitted
Stern-tube shaft	23–11 5/16	10	7	1	10 29/32	2–02	Lignumvitae
				1	10 27/32	3–08	Lignumvitae
Propeller shaft	28–10 3/16	10	7	1	10 29/32	2–07	Lignumvitae
				1	10 27/32	3–08	Lignumvitae

The shaft horsepower is obtained by a Gary-Cummings torsion-meter fitted in the first section of the line shafting.

Thrust bearings of the following characteristics are provided at the forward end of each turbine:

Collars on shaft, number	7
thickness, inches	1 1/16
space between, inches	3
outside diameter, inches	14⅜
inside diameter, inches	8¼
bearing surface, 6 collars, square inches	626.24
Thrust shoes, number	6
effective (thrust) surface, square inches (6)	388.44

PROPELLERS.

The propellers are three-bladed and turn outboard when going ahead. They are cast solid of manganese bronze, machined to a true helical driving surface, balanced and polished. The following are the propeller measurements:

	Starboard.	Port.
Diameter, inches	92½	92½
Pitch, inches	81⅞	81⅞
Area, helical, square inches	4,461.	4,449.
projected, square inches	4,005.	3,984.
disc, square inches	6,719.	6,719.
Ratio pitch to diameter, P/D	0.8851	0.8851
Height lower tip blade above keel, inches	3.662	
Immersion of upper tip blade at load draft light, inches	26.46	
Edge of propeller from C. L. ship, feet	2.	

MAIN AND AUGMENTER CONDENSERS.

There are two main condensers, each having an augmenter condenser. They are of oval section, with horizontally-curved tubes expanded into the tube sheets.

Data One Main Condenser.

Cooling surface, square feet	5,835
Tubes, length as fitted, feet and inches	15-4 to 15-9½
number	2,378
thickness, B.W.G.	18
outside diameter, inch	0⅝
Connections:	
Main exhaust, rectangular, square feet	15
Auxiliary exhaust, diameter, inches	7
Bleeder (one condenser only), diameter, inches	2½
Safety valve, diameter, inches	2
Air pump, wet suction, inches	8½
Air pump, dry suction (from augmenter), inches	9
Circulating water:	
Injection and discharge, diameter, inches	22

The augmenter condensers are of the Parsons type. Each has a cooling surface of 152 square feet.

MAIN AIR PUMPS AND CONNECTIONS.

There are two Blake, double, vertical, bucket, single-acting main-air pumps, located in the auxiliary room, amidships. Each pump has a 14-inch steam cylinder, two 28-inch water cylinders, 18-inch stroke, and a 12-inch suction from its own condenser, through a water seal. The water seals are cross connected. A make-up feed line from the reserve-feed tanks

joins each pump suction. The turbine drains are led to each pump. The pumps discharge separately through 10-inch discharge pipes to the first compartment of the feed and filter tank.

MAIN CIRCULATING PUMP AND ENGINE.

There are two main circulating pumps of the centrifugal type, each driven by a reciprocating engine, with steam cylinder 9-inch diameter by 8-inch stroke.

The engine is enclosed in a casing and connected to the forced-lubrication system, and has besides a small pump of its own driven from its shaft and supplying oil from a well.

FEED AND FILTER TANK.

The feed and filter tank, of about 800 gallons capacity, is located in the forward part of the auxiliary room amidships, over the main air pumps. It is divided into two parts by a horizontal partition. The upper part, of 300-gallons capacity, is further subdivided to furnish two filter chambers. The filter compartments may be by-passed.

MAIN FEED PUMPS.

Two Blake, vertical piston, double-acting, single main feed pumps are located in the port side of the engine room forward. Each pump has a 15-inch steam cylinder, 10-inch water cylinder, and 16-inch stroke. They have 5½-inch suctions from the feed and filter tanks, 3-inch suctions from the reserve-feed tanks, and 5½-inch suctions to water seals cross connected. They discharge through 4-inch feed line to the boilers.

AUXILIARY CONDENSER AND PUMPS.

An auxiliary condenser of 250 square feet cooling surface is located amidships in the after part of the auxiliary room. The shell is cylindrical, the tubes being curved in the vertical plane and expanded into the tube sheets. The tubes are supported at the middle and are protected from the auxiliary

exhaust by a perforated baffle plate. A Blake simplex, combined air and circulating pump, 6 inches × 8 inches × 8 inches × 7 inches, is provided.

FORCED-LUBRICATION SYSTEM.

A complete system of forced lubrication is provided for the main and thrust bearings and cruising engines.

BOILERS.

The boilers are of the Thornycroft type, arranged in pairs in two separate firerooms. They are equipped for fuel oil and each is fitted with seven Schutte-Koerting burners.

Data for One Boiler.

Designed working pressure, pounds	265
Test pressure, water, pounds	400
Heating surface, square feet	6,614
Combustion chamber, length, feet	9.9
width, feet	10.989
area, square feet	108.79
volume, cubic feet	622.175
Floor to top of steam drum, feet and inches	13–3$\frac{19}{32}$
Length, external, over steam drum, feet and inches	14–2$\frac{11}{32}$
Width, external, over casing, feet and inches	16–4$\frac{1}{2}$
Area, through cone opening, square feet	13.11
through flame opening, square feet	5.7
through uptake, square feet	19.4
smoke pipe, square feet	18.0
Diameter, steam drum, inches	42
water drums, inches	19
Tubes, number	176 and 2374
diameter, outside, inches	1$\frac{3}{4}$ and 1$\frac{1}{2}$
thickness, B.W.G	11 and 12

FUEL-OIL SYSTEM.

Fuel oil is carried in tanks outside the machinery spaces. There is a separate suction line from each of the forward tanks to a manifold in the forward fireroom. Similarly separate suction lines from each of the after tanks lead to a manifold in the auxiliary room. These two manifolds are connected

by a common suction main on which either the service or booster pumps can take suction.

Two booster pumps are provided, one in each fireroom. They have suctions from all tanks and deck connections, and discharges to the service pumps, suctions, storage tanks and deck-hose connection.

There are four service pumps, two in each fireroom. They have suctions from storage tanks, direct or via booster-pump charge, and discharge to the burners.

For raising steam when no pressure is available a hand pump, capable of supplying oil at 200 pounds pressure to two burners, is fitted in each boiler compartment.

There are eleven 2.5-mm. tip burners and tuyeres of the Schutte-Koerting type. Oil is supplied to each burner through $\frac{3}{8}$-inch lines at about 200 pounds pressure. The tuyeres are so arranged that the quantity of air through each may be independently regulated at each burner.

FORCED-DRAFT BLOWERS.

In each fireroom there are two Sturtevant, vertical, single-inlet cone fans, driven by Terry steam turbines. The fan and turbine are both on the same shaft, and are supported beneath the deck. Air is drawn down through the fireroom ventilators and driven out horizontally into the firerooms. The blowers are rated at 1,600 r.p.m., at which speed they deliver 33,000 cubic feet of air at 5 inches water pressure. The two after blowers have ducts leading to the engine room, so that warm air may be drawn from it.

FEED-WATER HEATERS.

Two Schutte-Koerting spirally-corrugated film feed-water heaters are located one in each fireroom. They are of the horizontal-pressure type and use auxiliary-exhaust steam as the heating agent. Both the main and auxiliary-feed pumps can discharge through them.

Data for One Feed-Water Heater.

Total heating surface, square feet	191.1
Outer tubes, number	32
thickness, B.W.G	13
outside diameter, inches	2.565
length as fitted, feet and inches	3–10
Inner tubes, number	32
thickness B.W.G	14
outer diameter, inches	2.166
length as fitted, feet and inches	4–5½
Shell diameter, inches	27⅝
Length over all, feet and inches	6–10⅝

MAIN STEAM PIPING.

Each boiler has a 7-inch main stop valve and steam line. The lines from boilers 1 and 2 unite in the forward fireroom in a 9½-inch line which passes along the port side to the engine room. The lines from 3 and 4 boilers unite in a 9½-inch line and pass along the starboard side to the engine room. Each line has a balanced slip joint in its fireroom and a bulkhead stop in the engine room. In the engine room there are 9½-inch throttle valves to ahead and astern turbines, 3½-inch throttle valves to the reciprocating engines, 2½-inch bleeder to port condenser, and steam connection to reversing engine. There is a 6½-inch cross connection, with slip joint and stop valve. Boiler and bulkhead stops are fitted with deck operating gear.

AUXILIARY STEAM PIPING.

There is a 3½-inch auxiliary stop valve on each boiler, which takes off from the main stop casting, and is fitted with deck-operating gear. In the forward fireroom branch lines run to the auxiliaries of that room, to forward deck machinery, heating system and galley, tank-steaming connections and whistle. The 3½-inch connections from the boilers of this room unite on the starboard side in a 4-inch line which runs aft into the after fireroom. Here there is a cut-out valve, and the line is joined by a 3½-inch line with cut-out valves from the two

boilers of this room. The lines of these two boilers in addition unite on the port side and continue aft in a 4-inch line, reducing to 3½ inches after passing the main circulating engines. The line on the starboard side reduces to 3½ inches at the same point, and continues aft to the auxiliary room, where it crosses over and joins the port line. There are two cut-out valves in this room, one on each side of the dynamos. The auxiliary steam line thus forms a loop embracing the after fireroom and the engine rooms, with a branch to the forward room.

AUXILIARY EXHAUST PIPING.

The auxiliary exhaust line, beginning on the port side of the forward fireroom as a 7-inch line, receives there the exhaust from all forward machinery. It passes aft on the port side, increasing to 8 inches as it receives the exhaust from the auxiliaries in the after fireroom. As it enters the engine room, it increases to 9 inches, and continues aft to the port after corner of the engine room. Here it divides into two 7-inch branches to each main condenser, and a 7-inch line which passes a spring-loaded valve to the auxiliary condenser. Steam from engine-room auxiliaries and auxiliary room, except dynamos and auxiliary condenser, unite in branches which join the 7-inch branch of the auxiliary exhaust to the starboard condenser.

In addition to the 7-inch exhaust connections to the main and auxiliary condensers, the exhaust line has 6½-inch connections to the main turbines at the 1st, 3d and 18th stages; 7-inch connections to the feed-water heaters, and 7½-inch escape pipe to atmosphere.

MAIN AND AUXILIARY FEED SYSTEM.

An 8-inch suction line runs from the feed tank with 5½-inch branches to the main and auxiliary feed pumps. The main feed pumps discharge through 4-inch discharges to a 6-inch main feed line which has 4-inch branches to each fireroom. These are so arranged that the feed water may pass by or

through the feed-water heaters, and enter the upper drum of the boilers. The auxiliary feed pumps can also discharge through or by the feed-water heaters.

The main feed pumps have in addition a 5½-inch suction to the water-seal cross-connection pipe, and 3-inch suctions to the reserve-feed tanks. The auxiliary feed pumps have 3-inch suctions to the reserve-feed tanks and 1½-inch hose connections for suction and discharge. Make-up connections join the main air-pump suctions. Overflow from the feed tanks may flow to bilge or return to the reserve tanks through a 4-inch line.

EVAPORATING AND DISTILLING PLANT.

There are two Schutte-Koerting single-effect evaporators in the auxiliary room, and above them, in the hatch, two distillers. The evaporators have a combined capacity of 3,750 gallons of potable water per 24 hours, and the distillers 2,500 gallons.

Evaporators take steam from the auxiliary steam line through 1½-inch lines and drain through a trap to the feed tank or to the starboard main condenser.

Evaporator feed is supplied by a 4½-inch × 6-inch × 6-inch Blake pump, which draws from the sea or from the overboard discharge of the distillers. It discharges through or by-passes a Department-type feed-water heater which uses the vapor from the evaporators as the heating agent. As steam forms in the evaporators it rises through 4-inch valves to a common 5½-inch vapor line to the distillers, from which it drains as water to a 50-gallon reservoir tank. A 3½-inch × 4-inch × 4-inch Blake pump pumps from this tank to the feed tank, fresh-water tanks or cofferdam. Vapor may also be admitted to the auxiliary exhaust line. Circulating water for the distillers is supplied by either fire and bilge pump.

Data for One Evaporator.

Type	Schutte-Koerting.
Diameter inside, inches	22
Height over all, feet and inches	5–9½
Rows of tubes, number	4

Tubes "U," number	32
diameter inside, inch	1
diameter outside, inches	1⅛
Water chamber	Bronze.
Heating surface, total, square feet	34.16

Data for One Distiller.

Type	Vertical, straight-tube, S. & K.
Diameter, inside, inches	11¾
Height over all, feet and inches	4–00½
Tubes, number	81
outside diameter, inch	00⅝
thickness, B.W.G., number	16
length as fitted, feet and inches	2–06⅝
Cooling surface, total, square feet	32

Data for Evaporator Feed Heater.

Type	"U" tube, Bureau.
Tubes, number	24
inside diameter, inches	1¼
thickness, B.W.G., number	14
Heating surface, square feet	18.76

ELECTRICAL PLANT.

There are two 10-kw. dynamos and switchboard on the port side of the auxiliary room. The generators, made by the General Electric Company, are driven by single-stage, three-row, Curtis turbines at 5,000 r.p.m., and deliver current at 125 volts. Lighting circuits from the switchboard lead to the machinery spaces and to panel boxes forward and aft in the living compartments. These boxes contain separate switches for each branch. Two power circuits with rheostats supply two 24-inch General Electric searchlights, one on the after deck house and the other above the bridge. A third power circuit carries current to the lathe. Call bells and the fuel-oil-filling alarm system are on a battery circuit.

There is a 2-kw. radio set, manufactured by the Garward Electrical Company. The motor generator transforms 125 volts D. C. to 110 A. C. at 550 cycles and makes 1,875 r.p.m.

MACHINERY WEIGHTS.

The following table gives the actual machinery weights:

Group.			Weight in pounds.
I.	Main engines:		
	Reciprocating engines, cylinders	12,825	
	Turbine casings, rotors, etc	137,114 =	149,939
II.	Shafting (line and propeller):		
	Reciprocating engines	5,870	
	Turbine engines	35,803 =	41,673
III.	Main engine framing and bearing:		
	Reciprocating engines	11,336	
	Turbine engines	10,062 =	21,398
IV.	Reciprocating parts (reciprocating engines)		3,496
V.	Main engine valve gear (reciprocating engines)		3,842
VI.	Main condensers, including augmenters		52,613
VII.	Main air and circulating pumps		27,519
VIII.	Propellers		8,753
IX.	Boilers		184,676
X.	Boiler fittings		47,586
XI.	Smoke pipes and uptakes, including guys		17,024
XII.	Steam and exhaust pipes and valves		54,136
XIII.	Suction and discharge pipes and valves		38,981
XIV.	Lagging and clothing		9,454
XV.	Flooring, grating, ladders, etc		11,862
XVI.	Auxiliaries		23,096
XVII.	Fittings and gear		16,997
XVIII.	Water		75,127
XIX.	Stores, tools and spare parts (exclusive of parts to navy yards, and parts of electrical plant)		11,695
XX.	Miscellaneous machinery, etc		60,582
XXI.	Electrical plant (exclusive of conduit and wiring, but including 404 pounds for its spare parts)		3,157
XXII.	Connections under steam engineering to other miscellaneous machinery		3,418
	Total weight in pounds		867,024
	Total weight in tons		387.06
	Contract weight in tons		387.45

TRIALS.

The contract required:

(*a*) A progressive trial over the measured-mile course at Delaware Breakwater, for standardizing the screws, extending from maximum speed down to a speed of 12 knots.

(*b*) A full-speed trial of four hours' duration in the open

sea in deep water, at the highest speed attainable, the average for the four hours not to be less than 29 knots. The speed to be determined by the average revolutions of the main shafts, according to the official standardization curve.

(*c*) A fuel-oil and water-consumption trial of four hours' duration in the open sea in deep water, at an average uniform speed of 24 knots, as nearly as possible. The trial to be conducted as nearly as possible to service cruising conditions.

(*d*) A fuel-oil and water-consumption trial of four hours' duration at 15½ knots, under conditions similar to the preceding trial, but with the cruising engine connected and in use.

TABLE I - STANDARDIZATION TRIAL DATA - JAN. 5, 1915 - U.S.S. DOWNES.

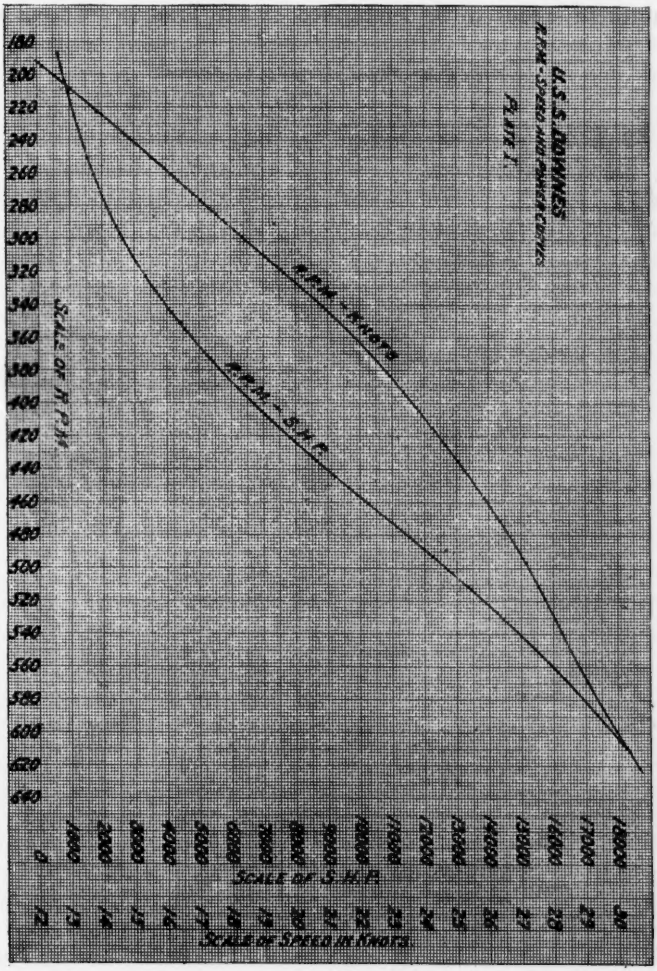
No. of Run	Time on Course. Mins.	Secs.	Speed in Knots.	Revs. to go One Knot. Star. Engine.	Port Engine.	Mean	R.P.M. Star. Engine.	Port Engine.	Mean.	S.H.P. Star. Engine.	Port Engine.	Total.
1	4	17.6	13.98	820.4	831.1	825.8	181.09	193.57	192.23	352	327	679
2	5	50.9	10.26	1157.9	1152.8	1155.4	138.04	197.17	197.61	392	360	752
3	4	18.5	13.93	823.4	823.5	823.5	191.07	191.09	191.08	352	336	688
MEAN OF GROUP			12.11						194.64			710
4	4	18.2	13.84	1091.9	1085.2	1088.6	253.76	252.23	253.01	843	767	1616
5	3	30.0	17.14	867.4	879.2	863.3	253.48	251.14	252.31	648	730	1378
6	4	17.9	13.96	1059.3	1050.0	1054.7	246.49	244.20	245.36	757	660	1417
MEAN OF GROUP			15.55						250.75			1547
7	3	6.9	19.92	923.7	918.5	921.6	297.53	296.32	296.89	1841	1241	2582
8	3	25.7	17.17	1042.1	1038.2	1040.2	298.31	297.18	297.75	1324	1225	2549
9	3	6.2	19.33	931.6	931.4	931.5	300.12	300.05	300.09	1373	1339	2712
MEAN OF GROUP			18.25						298.12			2559
10	2	55.9	20.47	1030.8	1032.4	1031.6	351.40	352.15	351.88	2280	2190	4470
11	2	41.3	22.32	947.0	947.5	947.3	352.05	352.22	352.14	2259	2166	4425
12	2	55.2	20.55	1028.1	1028.9	1028.5	352.07	352.35	352.21	2211	2120	4331
MEAN OF GROUP			21.62						352.09			4413
13	2	26.3	24.61	994.3	995.7	995.0	408.58	408.19	408.39	3598	3457	7055
14	2	31.6	23.78	1050.5	1057.8	1054.2	416.32	419.19	417.76	3845	3748	7593
15	2	25.9	24.67	1010.7	1014.3	1012.5	413.48	416.86	416.22	3858	3805	7663
MEAN OF GROUP			24.21						415.16			7996
16	2	17.6	26.16	1149.1	1148.9	1149.0	501.23	501.19	501.19	6399	6266	12665
17	2	6.1	28.55	1047.5	1051.5	1049.5	498.41	500.31	499.36	6431	6256	12687
18	2	19.9	25.73	1169.1	1169.8	1169.5	501.38	501.69	501.54	6304	6273	12777
MEAN OF GROUP			27.25						500.36			12709
20	2	12.3	27.21	1255.2	1256.3	1255.8	569.05	569.55	569.30	8353	8046	16399
21	1	56.5	30.90	1101.2	1103.7	1102.5	567.35	568.67	568.01	8289	8149	16438
22	2	12.7	27.13	1263.5	1265.4	1264.5	571.07	571.91	571.49	8304	8156	16460
MEAN OF GROUP			28.04						569.20			16434
23	1	53.3	31.77	1115.8	1122.2	1119.0	590.33	593.76	592.06	8824	8468	17292
24	2	10.6	27.61	1288.6	1294.5	1291.6	592.91	595.42	594.27	8503	8458	17401
25	1	52.8	31.91				593.53	597.49	595.51	8955	8783	17678
26	2	8.8	27.95	1319.6	1325.5	1322.6	596.86	599.53	598.20	8924	8793	17717
27	1	53.2	31.80	1074.3	1081.3	1077.8	595.17	599.04	597.11	8898	8786	17684
MEAN OF GROUP			29.815						595.65			17622

(*e*) An endurance trial of 20 hours' duration in the open sea at an average uniform speed of 15½ knots, as nearly as possible, following as closely as possible trial (*d*), with cruising engine connected and in use. Fuel-oil and water-consumption will not be measured on this trial, the purpose of which is to determine the reliability and endurance of the cruising engine.

(*f*) A fuel-oil and water-consumption trial of four hours' duration in the open sea, with the cruising engine connected

and in use, at an average uniform speed of 12 knots as nearly as possible.

(*g*) In addition to the above-enumerated trials the contract was amended to include a two-hours' trial at about 15½ knots, with the main turbines only in use, fuel-oil and water-consumption to be carefully measured on this trial.



Fuel-oil Consumption Guaranties.—The contractors guaranteed that the fuel-oil consumption per knot run for all purposes, including that necessary for all auxiliaries in use on the trials, would not exceed 700 pounds at guaranteed maximum speed, 460 pounds at 24 knots, 210 pounds at 15½ knots, and 175 pounds at 12 knots, the consumption of the fuel oil at these speeds to be determined by the Trial Board from a

TABLE II—TRIAL DATA—U.S.S. DOWNES.

	4 Hrs., Full Speed Trial (a)	4 Hrs., 24 Knot Trial, (c)	4 Hrs., 15½ Knot Trial, (d)	20 Hrs., 15½ Knot Trial, (e)	4 Hrs., 12 Knot Trial, (f)	2 Hrs., 15½ Knot Trial, (g)
Date of Trial	Jan. 6, 1915	Jan. 6, 1915	Jan. 7, 1915	Jan. 6&7, 1915	Jan. 7, 1915	Jan. 7, 1915
Speed in Knots	28.071	24.091	15.375		12.157	15.458
Draught, Mean on Trial, Feet and Inches	9-3⅝	9-3⅝	9-3⅝		9-10	9-10⅜
Displacement, Corresponding, Tons	1106	1106	1104		1112	1105
Slip of Propellers, Percent, Starb'd.	24.14	13.1	8.53		7.07	8.28
Port	24.39	13.29	7.27		7.92	7.66
Mean	24.27	13.17	7.9		7.5	7.97
Engines in Operation	Main Turbines		Main Turbines & Cruising Eng.			M. Turb.
Number of Boilers used	4		3		1	1
Burners used, (11 per Boiler)	42		33		11	11
Heating Surface used, Sq. Ft.	26,456	19,842	14,814		6614	6614
Sq. Ft. per S.H.P.	1.682	2.769	4.329		9.872	4.292
Cooling Surface, Sq. Ft., (Main Cond.), per S.H.P.	0.716	1.629	7.637		17.42	7.373
Pressures, (Average):						
Main Steam at Boilers, Pounds Gauge	251.81	246.46	245.63		240.0	241.88
" Eng. Room, " "	241.75	238.25	243.63		239.63	242.63
Ahead Steam Chest, Pounds Absolute	244.63	247.01	14.63		11.63	250.25
1st Stage, " "	101.9	91.53	19.13		9.65	18.63
4th " " "	33.88	18.3	4.82		3.25	6.31
18th " " "	22.75	9.42	2.5		1.94	3.21
Gland Steam, Pounds Gauge	3.97	1.96	1.93		1.82	2.30
Vacuum, Main Cond., Inches of Mercury	28.4	29.3	28.25		27.83	28.16
Barometer, Inches of Mercury	30.54	30.24	30.05		29.89	29.94
Auxiliary Steam, Pounds Gauge	254.25	248.75	244.38		240.0	243.5
Exhaust, " "	14.68	14.75	0.44		0.75	11.81
Main Feed at Pumps, " "	301.25	313.13	313.13		350.0	328.13
Forced Lubrication, " "	59.06	51.07	45.07		—	48.0
Fuel Oil, Pounds Gauge	208.0	136.88	153.88		130.0	165.0
Air Pressure in Fire Rooms, Inches of Water	5.46	4.085	4.38		3.4	4.75
Revolutions or Double Strokes per Min., (Average):						
Starb'd. Shaft	569.1	411.75	252.86		194.29	253.53
Port "	571.0	418.35	249.41		196.07	251.82
Mean	570.1	412.05	251.14		195.18	252.6
Main Air Pumps	(a) 25.72	(c) 20.88	(d) 13.5		(f) 9.0	(g) 15.75
Circulating Pumps	(a) 254.15	(c) 216.88	(d) 96.25		(f) 82.02	(g) 200.0
Feed Pumps	(a) 349.3	(c) 317.0	(d) 127.5		(f) 5.75	(g) 13.63
Forced Draft Blowers	(a) 435.8	(c) 1180.4	(d) 231.3		(f) 132.5	(g) 1500.0
Fuel Oil Service Pumps	(a) 32.43	(c) 15.94	(d) 9.38		(f) 6.13	(g) 11.5
Shaft Horse Power, (Average):						
Starb'd. Shaft	8324	3429	821 *		337 *	812
Port "	7984	3337	707 *		333 *	729
Total	16308	7166	1528 *		670 *	1541
Temperatures, Degrees F. (Average):						
Main Air Pump Discharge	78.89	71.07	56.19		47.5	53.57
Injection	47.56	46.75	34.75		34.5	38.5
Outboard Delivery	83.99	71.52	63.0		45.52	68.63
Outside Air	58.75	53.25	44.88		46.0	54.25
Engine Room	88.31	82.25	83.5		79.88	85.75
Auxiliary Room	84.06	78.5	83.63		83.25	88.38
Fire Rooms	82.97	83.24	67.63		71.88	69.25
Feed Water	157.53	165.1	173.5		191.88	168.5
Fuel Oil to Heaters	52.98	59.44	64.0		65.25	61.75
Burners	197.8	135.04	143.88		133.38	177.5
Smoke Pipe	569.2	506.88	430.0		413.75	675.63
Water Consumption:						
Pounds per Hour, Measured	301,231	152,860.5	40,745		25,881	55,476
Reserve Feed	2,764.5	1,064.0	1,174		1,959.25	349
Total Evaporated	303,987.5	153,924.5	41,921		27,840.25	56,218.5
per S.H.P.	18.471	21.331	26.666		38.628	36.13
Evap'd. per Sq. Ft. of H.S.	11.491	7.758	6.338		4.215	8.501
Lb. of Oil	12.31	13.34	13.13		13.84	12.76
Knot Run	18,457.871	6,388.764	2,691.903		2,293.398	3,590.816
Fuel Oil Consumption:						
B.T.U. per Pound	19,436	19,461	19,461		19,461	19,461
Sp. G. at 60°F.	0.8765	0.874	0.874		0.874	0.874
Pounds per Hour	24,465	10,966.8	3,103		1985	4363
Knot Run	841.56	455.24	199.56		163.88	278.96
At	19 (832	14 (450.5	13½ (198		12 (162	13½ (276
Guaranteed at	Kts. 700	Kts. 460.0	Kts. 210		Kts. 175	Kts. None
(a) Above or (b) Below Guarantee	(A) 132	(B) 9.5	(B) 12		(B) 13	
Hour per S.H.P.	1.5	1.531	2.031		2.963	2.831
Sq. Ft. of H.S.	0.925	0.553	0.469		0.3	0.66

NOTE:—FIGURES IN BRACKETS ( ) INDICATE NUMBER OF AUXILIARIES IN OPERATION.
*I.H.P., CRUISING ENGINES.—TRIAL (d), 363 STAR., 404 PORT, 767 TOTAL; TRIAL (f) 228 STAR., 191 PORT, 418 TOTAL.

ENDURANCE TEST OF MACHINERY. NO DATA TAKEN.

curve based on the rate of fuel oil consumed on trials (*b*), (*c*), (*d*) and (*f*).

Trial (*a*).—The standardization trial was successfully run on January 5, 1915, on the measured-mile course off Delaware Breakwater. In all 27 runs were made at various speeds. Run No. 19 being unsatisfactory was thrown out. From the data obtained it was found to require 567.6 r.p.m. of the propellers to attain the designed speed of 29 knots; 409.9 r.p.m. for 24 knots, 250 r.p.m. for 15½ knots, and 192.6 r.p.m. for 12 knots.

Table I contains the data from which the curves, Plate I, were plotted.

Trials (*b*), (*c*), (*d*), (*e*), (*f*) and (*g*) were conducted on the dates noted in Table II, which also gives the data obtained.

DESCRIPTION AND TRIALS OF STEAMSHIPS *GREAT NORTHERN* AND *NORTHERN PACIFIC*.

By W. B. Robins, Associate Member.

For the purpose of maintaining an express steamer service between San Francisco, Cal., and Astoria, Oregon, the Great Northern Pacific Steamship Company has had built by the Wm. Cramp & Sons Ship and Engine Building Company, Philadelphia, Pa., two ocean-going vessels, the *Great Northern* and the *Northern Pacific*.

These ships have a service speed of 23 knots, and at this speed will make the trip in from 25 to 26 hours, about the same time as it is required for the railway trip between San Francisco and Portland.

To both the naval architect and the traveling public these vessels represent a notable achievement in design and construction, being seaworthy, fast, and provided with every comfort and luxury for passengers. The hull is built on extremely fine lines, while at the same time ample space is provided for 856 passengers, 198 crew, about 2,185 tons of freight, and the 25,000 horsepower propelling plant, with its fuel and auxiliaries.

The passenger accommodations are divided into first, second and third classes, and the arrangement of staterooms, saloons, entrances, deck space, etc., provides for the comfort and convenience of all classes of passengers in all kinds of weather.

In order to get the necessary power for the high speed at which these vessels run, without encroaching upon the space required for passenger accommodation, the type of machinery adopted is a three-shaft Parsons turbine arrangement, together with specially designed oil-burning water-tube boilers.

The machinery arrangement is remarkable for its compactness without being anywhere so crowded as to interfere with efficient operation and attendance. Among the special features of the machinery plant are a system of Pioneer smoke indicators, and the Rites system of lubrication, using a large gravity tank for the oil supply to bearings.

GENERAL ARRANGEMENT.

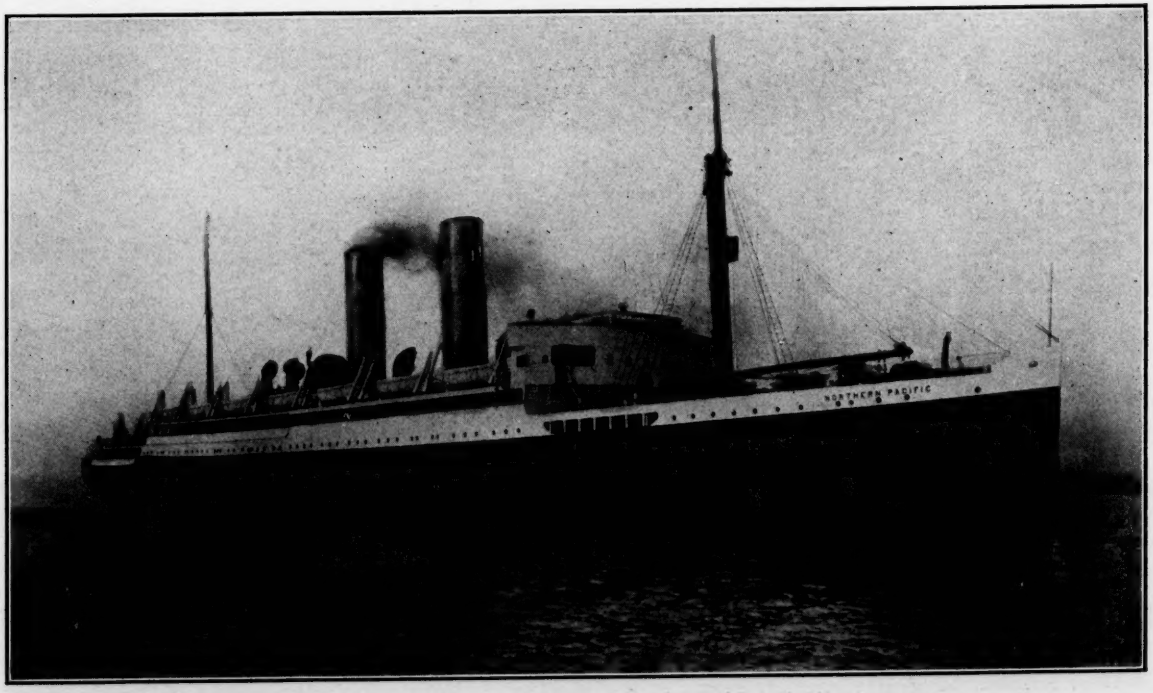
The vessels have straight stems, semi-elliptical sterns, two smoke stacks and two masts. There are six decks and a cellular double bottom, the latter being designed for carrying either fuel oil or water ballast. Beginning at the top, there are the A deck (promenade), B deck (bridge), C deck (shelter), D deck (upper) and E deck (second). Above the A deck there is a boat deck occupying about half the length of the ship. The boat deck is used for stowage of life boats and has a deck forward for officers' quarters, with pilot house and chart room above. The A deck consists principally of promenade space and has a long deck house containing the first-class observation room, smoking room and lobby, and a few staterooms. On this deck, a short distance forward of the deck house, there is a weather bulkhead completely across the ship and the sides are enclosed with sliding glass windows of the Laycock patent type for about 125 feet abaft this bulkhead, and a partly enclosed veranda is fitted up at the after end of the deck house. On the forward end of the A deck are the anchor windlass, 2 capstans and the winches for the forward cargo booms. The main part of the B deck is occupied by first-class staterooms. On the forward part of this deck are the third-class lavatories, dining room, and deck space; and at the after end are the second-class smoking room, lounge and deck space, the hand-steering gear and two capstans. The C deck is almost entirely occupied by staterooms; the first-class amidships, second-class aft and third-class forward. The quarters for the engineer's force are on this deck between the first and second-class passenger quarters. This deck also contains the seamen's quarters

forward, and quarters for the stewards, etc., aft. The steering engine is located on the after end of this deck. The D deck amidships is used for the first and second-class dining saloons, pantries and galleys. At the extreme forward end of this deck are third-class quarters and at the after end are stewards' quarters. There is cargo space on this deck both forward and abaft the main saloons, these spaces being fitted with large doors in the ship's sides. The E deck forward and abaft the machinery space is utilized principally as cargo space. On this deck, abaft the engine room, there is a cold-storage cargo compartment of about 19,460 cubic feet capacity. The hold consists principally of the machinery compartments and cargo space with fuel-oil tanks alongside the boiler rooms. The double bottom is divided in compartments for holding fuel oil and reserve feed water.

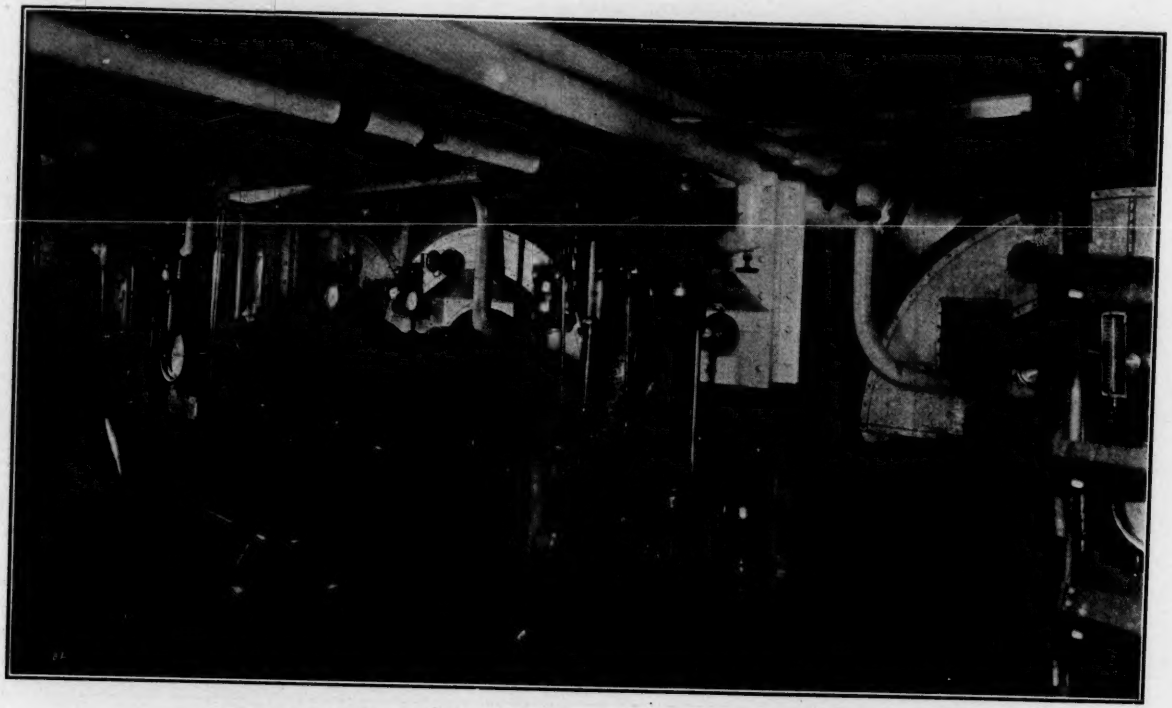
Hull Data.

Length overall, feet	524
between perpendiculars, feet	500
Beam, feet	63
Depth, molded to A deck, feet and inches	50–8
Draught, full load, feet	21
Displacement, tons	9,700
Deadweight carrying capacity, tons	2,185
Gross tonnage, tons	8,255
Capacity of fuel-oil compartments, tons	1,200
reserve-feed tanks, tons	112
drinking-water tanks, tons	102
Passengers, first-class	550
second-class	108
third-class	198
Total	856
Crew	198
Total persons on board	1,054

The stem is of rolled steel fitted to a cast-steel shoe piece, and the stern frame is cast steel. The rudder is solid cast steel with a forged-steel stock. The flat keel is double for three-fifths of its length. Besides the center keel there are three intercostal longitudinals on each side. Above the inner



S. S. "NORTHERN PACIFIC."



ENGINE ROOM.—S. S. "GREAT NORTHERN" AND "NORTHERN PACIFIC."

bottom there is a longitudinal bulkhead on each side extending from the forward boiler room bulkhead to the forward engine room bulkhead. The frames are spaced 30 inches amidships and 28 inches at the forward end. The A deck is completely plated for about one-half the length amidships, and this part of it is a strength deck. The B, C and D decks are completely plated.

Machinery Data.

Shaft horsepower, designed	22,000
Revolutions per minute, designed	350
Speed of ship, designed, knots	23
H.P. turbine, number of	1
length, overall, feet and inches	21–7½
diameter of rotor drum, feet and inches	5–8
number of stages	4
L.P. and astern turbines:	
Number of	2
Length overall, feet and inches	32–2
Diameter of rotor drum, L.P., feet and inches	7–8
Diameter of rotor drum, astern, feet and inches	6–7
Number of stages, L.P.	6
Number of stages, astern	4
Diameter of line shafting, inches	12¼
Propellers, solid manganese bronze, number of	3
number of blades, each	3
diameter, feet and inches	9–2
pitch, feet and inches	8–4
Main condensers, number of	2
cooling surface, each, square feet	13,046
total, square feet	26,092
number of tubes, each	6,018
diameter of tubes, inch	0.¾
thickness of tubes, B.W.G.	16
Boilers, number of	12
width overall, feet and inches	15–7⅜
length overall, feet and inches	11–8 7/16
height over steam drum, feet and inches	14–2½
diameter of steam drum, inches	54
number of tubes, each	969
diameter of tubes, inches	2
thickness of tubes, B.W.G.	9
heating surface, each, square feet	5,000
total, square feet	60,000
working pressure, pounds	220

The turbines are the Parsons type, arranged on three shafts, one high-pressure on the center shaft and one combined low-pressure and astern on each wing shaft. The turbine casings are of close-grained cast iron, the rotor drums are weldless-steel forgings. The drum wheels are steel castings and the rotor shafts are steel forgings. Lubrication of the turbine and main shaft bearings is by a gravity system. A tank of about 2,000 gallons, capacity is placed at height, which gives a pressure at the bearings of about 10 pounds per square inch. After the oil has passed through the bearings it flows to drain tanks from which it is returned to the gravity tank by two steam pumps, two coolers being fitted to lower the temperature of the oil.

The boilers are of the Mosher type, built by the Babcock & Wilcox Company, and are arranged in two compartments, six in each. The boilers are all set facing the center of the ship, leaving a firing space along the center line 11 feet 2½ inches wide from front to front of boilers. The total space occupied by the boilers is 130 feet x 41 feet 6 inches. At the forward end of the forward fireroom there is a donkey boiler 5 feet 9 inches, diameter, and 10 feet 7¾ inches long. The boilers use oil fuel, and operate under forced draft. The Koerting mechanical oil-burning system, made by the Schutte and Koerting Company, Philadelphia, Pa., is used. The principal features of this system are forcing the oil under high pressure through heaters and atomizing it mechanically in centrifugal spray-type burners without the use of steam or air as an atomizing agent, the air for combustion being furnished from the forced-draft pressure in the fireroom through an adjustable air register at each burner.

There are two smoke stacks, about 112 feet high above the base line. The Pioneer smoke indicator system is installed, which enables the density of smoke passing up the stacks to be determined in the fireroom. This system consists of an electric light in each smoke stack shining through a lens onto a selenium cell on the opposite side of the stack. The selenium cell is in an electric circuit, and a variation in the intensity of

the light changes the conductivity of the selenium, causing a variation in the current, this variation showing on an indicator in the fireroom.

The forced draft is operated on the closed-fireroom system, three 34-inch high-speed turbine-driven cone fans being fitted for each fireroom. These blowers are made by the B. F. Sturtevant Company, Hyde Park, Mass., and have a capacity of 30,000 cubic feet of air per minute against a pressure of 2 inches of water.

The following pumps were furnished by the Blake & Knowles Steam Pump Works:

2 Main air, vertical, twin-beam, single-acting, 14 x 32 x 21 inches.
1 Auxiliary air, vertical, featherweight simplex, 7½ x 14 x 10 inches.
4 Main feed, vertical simplex, 15½ x 9¾ x 24 inches.
2 Auxiliary feed, vertical duplex, 12 x 8½ x 12 inches.
2 Bilge and ballast, vertical duplex, 7½ x 9 x 10 inches.
2 Lubricating oil, vertical simplex, 6 x 8 x 12 inches.
4 Fuel-oil service, horizontal duplex, 7½ x 5 x 10 inches.
2 Fuel-oil transfer, vertical simplex, 6 x 8 x 12 inches.
2 Brine, vertical duplex, 5¼ x 5 x 5 inches.
1 Donkey boiler feed, horizontal simplex, 4½ x 2¾ x 6 inches.
1 Evaporator feed, horizontal simplex, 4½ x 2¾ x 6 inches.
2 Fresh-water, horizontal simplex, 5½ x 5½ x 7 inches.
1 Ice-water circulating, horizontal simplex, 3½ x 3½ x 4 inches.

Besides the above there are two 4-inch Worthington centrifugal sanitary pumps with a capacity of 500 gallons of salt water per minute against a head of 100 feet. These pumps are driven by a 20-horsepower Terry steam turbine, which runs at 2,750 revolutions per minute with initial steam pressure of 175 pounds and back pressure of 10 pounds. There are two 48-inch Reilly multicoil feed-water heaters, each having a capacity of 150,000 pounds of water per hour with a discharge temperature of 208 degrees F., with steam at 5 pounds pressure. There is also a Reilly multicoil Navy type evaporator, having a capacity of 25 tons of fresh water per 24 hours.

ELECTRIC PLANT.

Electric current is furnished by four 35 kw., 110-volt generators made by the Diehl Manufacturing Company. These generators are direct-connected to type "G M" Terry steam turbines, designed to operate with 200 pounds steam pressure and vacuum of 28 inches, but having sufficient power to deliver the rated output of the generators with 175 pounds steam pressure and a 26-inch vacuum; the normal running speed being 3,200 revolutions per minute. These generators supply current for lighting, etc., for operating five freight elevators and for operating a Willets-Bruce automatic whistle.

REFRIGERATING PLANT.

The refrigerating plant is the CO_2 type made by J. & E. Hall Company, Ltd. There are two machines having sufficient combined capacity to refrigerate the cargo compartment of 19,462 cubic feet and the provision compartments of 2,778 cubic feet, maintaining a temperature of 20 degrees F., when the outside temperature is 85 degrees F.

DECK EQUIPMENT.

The steam-steering gear is of the Williamson differential lever type with 14-inch x 14-inch cylinders, operated by telemotor from the bridge or direct at the engine. This gear, as well as the windless, capstans and deck winches, was made by the American Engineering Company, Philadelphia. There are three bower anchors of 9,000 pounds each; one stream anchor of 2,500 pounds, and one kedge of 1,200 pounds. There are two 2½-inch chain cables, each 150 fathoms long, and one 7-inch hawser, 120 fathoms long, for the stream anchor.

The following small boats are furnished: Thirteen metallic life boats, 28 feet x 9 feet 4 inches x 4 feet 1½ inches, 64 persons each; four collapsible life boats, 29 feet x 9 feet 10 inches

x 3 feet, 54 persons each; one metallic working boat, 22 feet x 6 feet x 2 feet 3 inches. The boats are handled by Horton sheath davits.

FIRE-ALARM SYSTEM.

The Aero Automatic Fire Alarm System is installed throughout the principal compartments of the ship. Each unit of this system consists of a small copper tube leading from a switchboard through its compartment and back to the switchboard. Any rapid rise of temperature in the compartment causes expansion of the air in the tube which operates a sensitive diaphragm at the switchboard, closing an electric circuit which rings a fire bell in the engine room and a buzzer at the switchboard. There are three divisions of the system, the three switchboards being located forward, amidships and aft on the C deck. There are two main annunciator boards, one in the engine room and one in the pilot house. The annunciators are in the form of diagrams of the ship, and the alarm shows in what part of the ship and on which deck the fire is located. The division of the system into three systems reduces the danger of panic, as an alarm is given only on the switchboard for its part of the ship, in the engine room and in the pilot house, thus giving an opportunity to put out the fire without alarming the passengers in the other parts of the ship.

TRIALS.

The contract required the following:

(*a*) A progressive standardization trial over the measured mile to standardize the propellers.

(*b*) A 12-hour trial at 20 knots; the revolutions to be taken from the revolution curve obtained by (*a*); and further, on this trial ten of the twelve boilers shall be in use, and four of the six forced-draft blowers.

The S. S. *Great Northern* was the first vessel ready for trial. It was decided to standardize over the measured mile at Rockland, Maine.

The standardization trial was conducted in December, 1914,

and the curve in Plate I obtained. Immediately after the three high-speed runs, the vessel began her 12-hour, 23-knot run. The speed was easily maintained and the performance of the machinery was excellent. Data is given in Table I.

En route to Rockland, the *Great Northern* encountered a heavy N. E. gale, and the owners were given an opportunity to ascertain her sea-going qualities. Accordingly the vessel was run at various speeds, up to full speed, with the sea ahead, on the bow and beam. The maximum pitch was 4 degrees to 5 degrees, and the maximum average roll 18 degrees to 20 degrees. The vessel was remarkably steady and very dry forward.

The S. S. *Northern Pacific* underwent her trials in February, 1915, and her performance was most excellent. Her average speed was slightly higher. The data is given in Table II.

One of the most excellent qualities of these vessels is the entire absence of vibration at high speed in the passenger quarters. There is only a slight vibration over the propellers.

In the engine design of the vessel there are several noteworthy ideas.

The arrangement in the engine room and firerooms is par-excellent. In the engine room it is possible to stand on the throttle platform and observe the operation of every piece of auxiliary machinery. This feature is extremely valuable to merchant service, where the maximum results must be obtained from the minimum cost.

The arrangement of the oil coolers under the sanitary pumps, the foundations for the pumps being cast with the cooler castings, and the water passing through them to the sanitary system, is one deserving of attention as to what can be saved in weight and space with a gain in efficiency.

The turbine forced-lubrication system is about the same as the standard Parsons system, except the pressure is maintained by the height of the supply tank in the engine-room hatch. There is an oil-level sight glass on the working platform and also a float in the tank, so that as soon as the oil drops to a

12 HOUR - 23 KNOT RUN. DEC. 12, 1914. 8:10 A.M. TO 8:10 P.M. FROM ROCKLAND

ENGINE ROOM DATA.

	TIME		A.M. 8.10 TO 9.30	A.M. 10.30	A.M. 11.30	P.M. 12.30	P.M. 1.30	P.M. 2.30	P.M. 3.30	P.M. 4.30	P.M. 5.30	P.M. 6.30	P.M. 7.30	P.M. 8.10	AVERAGES 10HR 40 MIN 9.30 A.M. TO 8.10 P.M.
	R.P.M. FOR PERIOD	P.	363.7	5383	3446	3467	3424	3471	3492	350.5	3488	3435	3522	352.	347.0
	DO.	C.	356.4	3205	3253	3279	3287	3286	3301	331.3	329.1	3278	3334	3334	328.5
	DO.	S.	358.0	3339	3387	3414	3426	3424	3445	3457	3439	3379	3475	3475	3421
	AVERAGE		352.7	3309	336	3387	3396	3394	3413	342.5	3487	336.4	3444	3442	3382
	RPM FOR TORSION METER CARDS	P.													3417
	DO.	C.													3303
	DO.	S.													3424
	AVERAGE														3401
	S.H.P.	P.													7290
	DO.	C.													7120
	DO.	S.													7070
	TOTAL														21480
PRESSURES LBS. SQ. IN. GAUGE	MAIN STEAM	P.		205	203	208	203	201	210	208	196	210	209	209	205
	DO.	S.		202	202	205	202	201	209	206	196	208	208	208	204
	MAIN STEAM ENGINE ROOM			198	198	202	196	198	203	200	190	202	202	201	199
	MAIN H.P. AHEAD			160	162	165	162	160	169	167	158	172	171	162	164
	L.P. AHEAD	P.		19	25	26	20	20	22	21	19	22	22	22	22
	DO.	S.		24	24	25	25	24	26	25	23	26	26	26	25
	VACUUM MAIN CONDNSR	P.		29.3	29.3	29.3	28.9	28.9	28.8	28.7	28.8	28.8	28.7	28.6	28.8
	DO.	S.		29.6	29.6	29.5	29.5	29.5	29.3	29.3	29.	29.4	29.2	29.2	29.4
	AUXILIARY STEAM			205	208	210	205	202	214	210	199	212	212	212	208
	" EXHAUST			4	4	4	3	4	4	3.5	3.5	4	4	4	3.8
	STEAM TO GLANDS			1.5	1.5	1.75	1.5	1.5	2	1.9	1.	2.25	2.25	2.25	1.8
	EXHAUST TO FEED HEATER	P.		6	1	1	1	1.25	2	2	2	1	.5		1.8
	DO.	S.		4	—	—	3	—	.5	.5	—	—	—		.8
	FEED PUMP DISCHARGE	P.		270	265	260	265	270	265	270	270	270	270		267
	DO.	S.		260	265	265	270	265	270	270	270	270	265		267
	FEED LINE			268	253	262	255	253	269	261	254	265	255	258	259
TEMPERATURES F.	TEMPERATURE OF INJECTION			45.	46	46	46	46	46	47	47.5	48	48.5		46.6
	OVERBOARD DISCHARGE	P.		73.5	74	74	74.5	74.5	74.5	75	76	77	77.5		75
	DO.	S.		74	74	74	74	74	74	76	76	76	78		75
	AIR PUMP DISCHARGE	P.		56.5	55	55	56	56	56	56	55	59	59		56.4
	DO.	S.		58	57	54	57	57	57	58	55	60	61		57.4
	FEED TO HEATER	P.		83	74	76	75	76	77	80	73	86	82		78.2
	DO.	S.		84	86	77	82	84	80	83	74	88	86		82.4
	FEED FROM HEATER	P.		218	184	180	198	188	197	206	180	196	190		192.7
	DO.	S.		176	208	188	184	198	194	200	170	187	180		187.5
	WORKING PLATFORM			91.	91.5	94	94	94	93	88	86	87	78	75	88.3
RPMS & R.P.M.	R.P.M. CENTRIFUGAL PUMP	P.		123	125	127	126	135	126	126	128	126	129		126.1
	DO.	S.		122	120	128	127	128	126	126	127	126	122		125.2
	AIR PUMP	P.		19	19.5	19.5	13	12	13.5	12.5	11	12.5	12		14.5
	DO.	S.		13	12	12	12	12	10	10	10	10	10.		11.1
	FEED PUMP	O.P.		9.8	14.3	15.2	8.7	3.9	6.4	6.4	6.6	6.2	6.4		8.4
	DO.	I.P.		4.9	9.7	10.5	14.1	16.8	17.2	18.1	18.4	17.1	17.6		14.4
	DO.	I.S.		12	9	12	12	15.3	17.1	17.8	17.9	18.0	19.3		15.0
	DO.	O.S.		19.3	14.6	14.2	14.5	13.0	8.5	8.9	8.9	8.9	9.7		12.1
	BILGE & BALLAST PUMP	P.		30	44	41	—	51	39	6.8	73	62	54		50.2
	DO.	S.		—	—	—	—	—	—	—	—	—	—		—
	LUBRICATING OIL PUMP	P.		6	7	7	6.5	6.5	6	6	6	10	9.5		7
	DO.	S.		12	12	12	12	12	13	13	14.5	12	12		12.2
LUBRICATING OIL PRESSURES LBS. SQ. IN.	DISCH. LUB. OIL PUMP	P.		27	33	34	32	33	32	32	31	36	33		32.5
	DO.	S.		33	34	35	33	33	35	35.5	32.5	37	36		34.2
	OIL THRUST BEARING	P.		10	9	10	10	10	9.5	9.5	9	10	9		9.6
	DO.	C.		7.5	8	7.5	7	7	8	7.5	7	8	7.5		7.5
	DO.	S.		9	8.5	8.5	9	8.5	8.5	8	8.5	9	8		8.5
	OIL FORD MAIN BEARING	P.		9	10	9.5	9	9	10	9.5	9	10	10		9.5
	DO.	C.		9.5	9.5	10	9.5	9	9	9	9	10	9.5		9.4
	DO.	S.		10	9	10	9	9	9	9	9	10	9		9.3
	OIL AFTER " "	P.		9.5	9	9	9	9	8.5	8.5	9	9	9		8.9
	" " " " "	C.		10.5	10.5	10	11	10	10	10	9.5	10	10		10.2
	" " " " "	S.		7	7	7	7	7	7	7	7	7.5	7.5		7.1
TEMPERATURES F.	OIL DISCH. THRUST BEARING	P.		98	100	100	101	100	100	100	100	100	99		99.8
	DO.	C.		92	94	94	94	94	94	94	94	95	93		93.8
	DO.	S.		101	102	103	102	103	103	103	103	103	101		102.8
	" " FORD MAIN "	P.		104	105	106	106	106	106	103	105	107	104		105.4
	DO.	C.		97	98	99	98	98	98	98	98	99	98		98.1
	DO.	S.		106	105	106	106	106	105	105	105	104	104		105.1
	" " AFTER MAIN "	P.		103	106	107	106	106	105	105	105	105	103		105.1
	DO.	C.		97	99	100	97	96	95	95	96	100	96		97.1
	DO.	S.		106	106	107	107	107	107	107	108	107	104		106.6
	LUB. OIL DISCH. FROM COOLER			82	83	83	82	82	82	82	85	84	84		82.7
	PRESSURE OF OIL SYSTEM			9	9	9.2	9	9	8.5	8.5	8.25	8.5	9	9	8.8

8.10 A.M. BEGINNING 1ST. STANDARDIZATION RUN AT 23 KNOTS AND ALSO
BEGINNING OF 12 HOUR 23 KNOT RUN.
9.20 A.M. FINISH 3RD & LAST STANDARDIZATION RUN AT MAXIMUM SPEED.
RECORDS FOR OIL, ETC, BEGIN AT 9.30 A.M.

TABL

CKLAND TO 23 MILES WEST OF NANTUCKET SHOAL LIGHTSHIP.
BOILER ROOM DATA.

AVERAGES FOR 10HR-40 MIN 10AM-8.40PM	12 HRS. 10AM-8.40PM		TIME	A.M. 10.00	A.M. 10.40	A.M. 11.30	P.M. 12.30	P.M. 1.30	P.M. 2.30	P.M. 3.30	P.M. 4.30	P.M. 6.00	P.M. 7.30	AVERAGE FROM 10AM-7.30PM.
547.0	548.9	BOILER PRESSURES FORWARD BOILER ROOM	P. FORWARD	197 MT	—	—	—	—	—	—	—	—	—	
528.5	529.4		S. "	223 MT	—	—	—	—	—	—	—	225 P	225 X	225
542.1	544.0		P. MIDDLE	202	210	208	212	212	212	216	220	220	220	213
539.2	540.3		S. "	203	210	210	215	212	212	215	218	235	230	214
542.7			P. AFT	203	208	210	215	213	214	218	220	220	218	214
530.3			S. "	202	208	212	217	215	212	218	218	220	220	214
542.4		BOILER PRESSURES AFT BOILER ROOM	P. FORWARD	205	205	215	215	215	218	220	220	200	220	215
540.1			S. "	205	205	210	210	210	215	218	218	200	220	211
729.0			P. MIDDLE	205	205	210	210	210	210	218	215	200	220	210
712.0			S. "	205	200	210	210	210	210	218	215	200	220	210
707.0			P. AFT	210	210	215	218	218	215	220	220	OUT P	— X	216 X
214.80			S. "	210	205	205	215	210	215	220	218	200	220	213
205		D.S. & R.P.M. BLOWERS AFT PORT BOILER ROOM	R.P.M. FORWARD	1390	1400	1400	1400	1400	1400	1475	1420	1350	1350	1390
204			" MIDDLE	OUT	—	—	—	—	—	—	—	1350	1375	1363
199			" AFT	1300	1400	1400	1400	1400	1400	1475	1430	1350	1350	1390
164			" FORWARD	1240	1400	1400	1400	1400	1400	1400	1400	1350	1350	1374
92			" MIDDLE	1240	1400	1400	1400	1400	1400	1400	1400	1350	1350	1374
25			" AFT	1220	1400	1400	1440	1420	1420	1430	1440	1400	1400	1401
289			AUX. FEED PUMP F.	—	—	—	—	—	—	—	—	—	—	—
294			DO. A.	—	—	—	—	—	—	—	—	—	—	—
208			FUEL OIL PUMP TOP F.	—	—	—	—	—	—	—	—	—	—	—
8.8			DO. BOTTOM F.	9	8	6	6	5	5	5	8	6	7	6.3
1.8			DO. TOP A.	—	—	—	—	—	—	—	—	—	—	—
1.8			DO. BOTTOM A.	9	16	12	15	13	14	17	16	14	17	14.3
8		PRESSURES-GAGE	AUX. STEAM F.	215		215	215	213	213	220	—	—	—	216
267			DO. A.	200	205	218	218	215	218	222	220	215	225	216
267			AIR PRESS. ROOM & UPTAKE F.	1.9	2.3	2.3	2.3	2.2	2.2	2.3	2.5	2.35	2.5	2.24
259			" IN INCHES WATER A.	2.0	2.1	2.3	2.3	2.3	2.35	2.35	2.35	2.4	2.5	2.35
46.6			MAIN FEED F.	—	—	—	—	—	—	—	—	—	—	—
75			DO. A.	—	—	—	—	—	—	—	—	—	—	—
73			AUX. FEED F.	270	250	250	250	250	250	250	—	—	—	253
564			DO. A.	—	—	—	—	—	—	—	—	—	—	—
57.4			OIL PUMP DISCH. F.	155	140	190	190	190	175	175	185	180	210	181
78.2			DO. A.	160	160	200	200	190	190	190	190	200	210	189
82.4			OIL SYSTEM F.	155	140	190	190	175	175	185	170	200	200	178
192.7			DO. A.	160	150	190	190	180	180	180	180	190	200	180
187.5		TEMP. PERA-TURES F°	OIL TO BURNERS F.	194	178	196	192	172	180	180	170	210	196	177.8
88.3			DO. A.	220	215	190	190	186	190	185	185	180	210	180
1261			BASE OF STACK F.	490	490	490	480	490	470	470	480	490	500	485
1252			DO. A.	700	680	680	680	680	670	680	670	690	700	675
14.5		NO. OF BURNERS SIZE OF TIPS 1908 AFT BOILER ROOM FORWARD BOILER ROOM	P. FORWARD	OUT	—	—	—	—	—	—	—	—	—	—
11.1			S. "	OUT	—	—	—	—	—	—	—	X 4	X 8	X 6
8.4			P. MIDDLE	7	7	7	8	8	7	8	8	7	8	7.5
14.4			S. "	7	7	6	7	7	7	8	8	7	8	7.3
15.0			P. AFT	7	7	8	7	7	8	8	7	7	7	7.3
19.1			S. "	7	7	7	8	8	8	8	8	7	7	7.3
302			P. FORWARD	8	8	7	7	7	8	7	7	8	6	7.3
—			S. "	7	8	7	7	7	8	7	7	8	6	7.2
7			P. MIDDLE	8	8	7	7	7	7	8	8	8	8	7.6
12.2			S. "	8	8	7	7	7	7	8	8	8	8	7.6
32.5			P. AFT	8	8	8	8		7	8	8	X OUT	X —	X 7.9
34.2			S. "	8	8	8	8		7	8	8	8	8	7.9
9.6		AIR PRESSURE IN INCHES OF WATER BETWEEN	BOILER ROOM & ATMOS.			1.8	1.8	1.8	1.8	1.85	1.8	2	2	1.86
7.5			DO. UPTAKE & FURNACE											2.3
8.5			DO. FURNACE & AROUND BURNERS											1.5
9.5			DO.											5.70.8
9.4														
9.3														
8.9														
10.2														
7.1														
99.8														
93.8														
102.8														
105.4														
98.1														
105.1														
104.1														
97.1														
106.6														
82.7														
8.8														

TEMPERATURE AT WORKING LEVEL = 60°F
" GRATING = 80°F
SMOKE USUALLY VERY LIGHT.
WATER CARRIED AT HALF GLASS.
ONE OIL HEATER IN USE IN EACH ROOM.

TEMP. OF GASES IN TUBE NESTS AT BAFFLE DIVISIONS. { BACK 1100° TO 1150° F. MIDDLE 950° TO 1000° F. FRONT 800° TO 850° F.

UPTAKE TEMP. WITH ALL BOILERS IN = 700°F. EXCEPT AT EXTREME ENDS WHERE DROP OF 200°F. WAS USUAL. A DROP OF ABOUT 150°F. WAS RECORDED WHEN ONE OR MORE BOILERS WERE CUT OUT DUE TO MIXTURE OF COLD AIR FROM IDLE BOILERS. X-PORT AFT BOILER CUT OUT & STRBD. FORD. BOILER PUT IN ITS PLACE [TUBE BLEW]

S.S. GREAT NORTHERN

TABLE I.

ENGINE ROOM DATA.

			4-HOUR 23 KNOT TRIAL									STANDARD DELAWARE	
	TIME		A.M. 10.50	A.M. 11.20	A.M. 11.50	P.M. 12.20	P.M. 12.50	P.M. 1.20	P.M. 1.50	P.M. 2.20	AVERAGE	P.M. 5.10	P.M. 5.25
	R.P.M. FOR PERIOD	P.									349.4		
	DO.	C.									328.9		
	DO	S.									346.4		
	AVERAGE										341.6		
	R.P.M. FOR TORSION METER CARDS	P.									350.0		
	DO.	C.									328.9		
	DO	S.									347.0		
	AVERAGE										342.0		
	S.H.P.	P.									8100		
	DO.	C.									7480		
	DO	S.									7320		
	TOTAL										22900		
PRESSURES LBS. PER SQ. INCH	MAIN STEAM LINE	P.	205	202	199	196	193	200	200	202	200	203	203
	DO.	S.	205	203	200	196	194	200	200	205	200	204	204
	MAIN STEAM ENGINE ROOM		195	190	190	187	185	192	190	193	190	195	195
	MAIN H.P. AHEAD		174	182	178	175	174	181	180	184	178	182	187
	L.P. AHEAD	P.	23.8	25.5	25	24.5	24	26+	26	26.6	25.7	27	27.5
	DO	S.	25	27	26.8	26	26		27.5	28	26.6	27	28.75
	VACUUM MAIN CONDNSR.	P.	28.6	28.5	28.5	28.2	28.3	28.4	28.6	28.6	28.4	28.75	28.75
	DO.	S.	28.7	28.6	28.5	28.5	28.5	28.5	28.5	28.7	28.5		
	AUXILIARY STEAM		210	206	204	200	199	205	205	206	204	207	209
	DO. EXHAUST		9	6.5	6	4	4	2.5	4	3.5	4.8	3	3.5
	STEAM TO GLANDS		3	2.5	2.7	1.	0	0+	.3	1.	1.3	1.	.5
	EXHAUST TO FEED HEATER	P.	4		5				2	5	4		
	DO	S.	6		6.5				5	2	4.9		
	FEED PUMP DISCHARGE	P.	235		240				250	248	241		
	DO.	S.	225		225				250	250	235		
	FEED LINE		235	220	213	215	210	225	230	235	223	235	240
TEMPERATURES F.	TEMPERATURE OF INJECTION		45		46				46	46	46		
	OVERBOARD DISCHARGE	P.	76		78				77	78	77.2		
	DO.	S.	75		74				74	75	74.5		
	AIR PUMP DISCHARGE	P.	58		59				58	58	55.7		
	DO.	S.	58		58				60	60	59		
	FEED TO HEATER	P.											
	DO.	S.	73		74				80	82	79.7		
	FEED FROM HEATER	P.	194	188	180	172	154	160	174	184	186	192	184
	DO.	S.	183	171	162	148	157	196	190	204	164	156	168
	WORKING PLATFORM		97	93	93	95	85	74	92	94	90.1	100	100
D.S. & R.P.M.	R.P.M. CENTRIFUGAL PUMP	P.	116		114				109	109	112		
	DO.	S.	118		120				120	119	119		
	AIR PUMP	P.	18		17				18	18	17.7		
	DO.	S.	20.5		21				19	21	20.4		
	FEED PUMP	O.P.	11		8				11	12	10.3		
	DO.	I.P.	16		16				12	14	14.6		
	DO.	I.S.	8		12				8	11	9.7		
	DO.	O.S.	13		13				15	12	13.2		
	BILGE & BALLAST PUMP	P.											
	DO.	S.	52		54				48	52	51.5		
LUBRICATING OIL	LUBRICATING OIL PUMP	P.	21		23				23	23	21.7		
	DO.	S.											
TEMPERATURES	DISCH. LUB. OIL PUMP	P.	40		40				41	42	40.7		
	DO.	S.											
PRESSURES 45 LBS.	PRESSURE OF OIL SYSTEM		6.1	6.2	6.2	6.2	6.3	6.5	6.5	6.5	6.3	6.5	6.5
	OIL THRUST BEARING	P.	7		7				7.5	7.5	7.2		
	DO.	C.	7		7.5				7.	7.5	7.2		
	DO	S.	7		8				6.5	7.	7.1		
	OIL FORWARD MAIN BEARING	P.	7.5		8				8.5	8.7	8.2		
	DO.	C.	7.5		8				7.	7.5	7.5		
	DO.	S.	8		8				8.	8.25	8.1		
	OIL AFTER DO.	P.	8.5		8.5				9	8.5	8.6		
	DO. DO.	C.	4.5		4.5				7	7.	5.7		
	DO. DO.	S.	8		8				8.5	8.5	8.2		
	OIL DISCH. THRUST BEARING	P.	97		98				97	98	97.5		
	DO.	C.	94		94				94	95	94.2		
	DO.	S.	97		98				98	99	98		
	DO. FOR'D MAIN DO.	P.	100		100				100	101	100		
	DO.	C.	98		98				98	99	98.2		
	DO.	S.	102		102				102	103	102		
	DO. AFTER DO.	P.	108		109				108	109	108		
	DO.	C.	94		95				96	96	95.2		
	DO.	S.	108		108				108	109	108.2		
	LUB. OIL DISCH. FROM COOLER		84		84				84	84	84		

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TABLE I

BOILER ROOM DATA.

4-HOUR 23 KNOT TRIAL.

STANDARDIZATION AT DELAWARE BREAKWATER.											
P.M. 5.10	P.M. 5.25	P.M. 5.45	AVERAGE		TIME	A.M. 11.00	A.M. 11.30	P.M. 1.00	P.M. 1.30	P.M. 2.30	AVERAGE
				BOILER PRESSURES. FORWARD BOILER ROOM.	P. FORWARD	212	212	216	213	215	214
					S. "	213	215	212	213	217	214
					P. MIDDLE	212	213	210	213	217	213
					S. "	217	217	215	217	218	217
					P. AFT	217	216	212	218	217	216
					S. "	216	216	214	217	217	216
				AFT BOILER ROOM	P. FORWARD	210	208	212	208	210	210
					S. "	210	207	212	208	213	210
					P. MIDDLE	212	213	215	212	213	213
					S. "	208	208	210	208	210	209
					P. AFT	207	208	208	205	208	207
					S. "	205	207	207	203	208	206
203	203	203	203	R.P.M. BLOWERS AFT BOILER ROOM. FORD BOILER ROOM.	R.P.M. FORWARD	1240	1300	1320	1300	1280	1292
204	204	204	204		" MIDDLE	1340	1340	1350	1340	1350	1344
195	195	195	195		" AFT	1260	1300	1280	1300	1320	1260
182	187	187	185.3		" FORWARD	1460	1480	1480	1480	1420	1264
27	27.5	28	27.5		" MIDDLE						
27	28.75	29	28.2		" AFT	1240	1400	1440	1460	1480	1404
28.75	28.75	28.75	28.75	D.S. & P. PRESSURES-GAGE.	AUX. FEED PUMP F.						
					DO. A.						
207	209	209	2083		FUEL OIL PUMP TOP F.						
3	3.2	4.9	3.8		DO. BOTTOM F.	5.5	6	9	9	8	7.5
1.	.5	.5	.6		DO. TOP A.						
					DO. BOTTOM A.	14	13	11	11	11	12
					AUX. STEAM F.	213	215	212	215	215	214
					DO. A.	213	215	213	218	218	216
					AIR PRESS. ROOM & UPTAKE F.	2.1	2.2	2.3	2.5	2.5	2.3
235	240	260	343.3		DO. IN INS. OF WATER A.	2	2.2	2.2	2.3	2.3	2.2
					MAIN FEED F.	230	230	240	240	250	258
					DO. A.	230	230	240	230	250	232
					AUX. FEED F.						
					DO. A.						
					OIL PUMP DISCH. F.	190	190	180	190	190	188
					DO. A.	190	180	180	190	190	186
					OIL SYSTEM F.	190	190	180	180	190	186
192	184	212	196		DO. A.	190	180	180	190	190	186
156	168	199	174.3	TEMPERATURES F°	OIL TO BURNERS F.						
100	100	100	100		DO. A.	52	52	53	53	53	52.6
					BASE OF STACK F.	550	610	520	500	480	512
					DO. A.	510	510	510	510	510	510
				No OF BURNERS SIZE OF TIPS. AFT BOILER ROOM. FORD BOILER ROOM.	P. FORWARD	5	5	5	6	6	5.4
					S. "	5	5	5	5	5	4.6
					P. MIDDLE	5	6	6	6	6	5.8
					S. "	5	6	6	5	6	5.4
					P. AFT	6	6	6	6	6	6
					S. "	6	6	6	6	6	6
					P. FORWARD	6	4	4	5	4	4.6
					S. "	5	4	4	5	4	4.4
					P. MIDDLE	5	5	5	5	5	5
					S. "	5	5	5	4.5	5	4.9
					P. AFT	5	5	5	5	5	5
					S. "	5	5	5	5	5	5
6.5	6.5	6	6.3	AIR PRESSURE IN INCHES OF WATER BETWEEN	BOILER ROOM & ATMOSPHERE	1.45	1.3	1.05	1.6	1.9	1.5
					DO. UPTAKE						
					DO. FURNACE						
					DO. & AROUND BURNERS						

4-HOUR 23 KNOT. FEB. 28, 1915, FROM 10.50 A.M. TO 2.50 P.M. AT SEA OFF DELAWARE CAPES. WEATHER - CLEAR AND BRIGHT. SMALL CHOPPY SEA. BAROMETER 29.93 TEMPERATURE OUTSIDE AIR 37° F. SPEED (FROM GREAT NORTHERN CURVE) 23.17 KNOTS. R.P.M. 341.6 S.H.P. 22900. DISPLACEMENT 9500 TONS.

STANDARDIZATION TRIALS. 3 RUNS OVER DELAWARE BREAKWATER COURSE FEB. 28, 1915 FROM 5.10 P.M. TO 5.50 P.M. WEATER CLEAR & BRIGHT SMOOTH SEA. AVERAGE SPEED, 23.28 KNOTS. R.P.M. 347.5

THIS TRIAL WAS MADE OVER THE DELAWARE BREAKWATER COURSE AND WAS ONLY FOR THE PURPOSE OF DETERMINING DIFFERENCE IN RELATIVE RESULTS BETWEEN THE DELAWARE COURSE & THE ROCKLAND COURSE ON WHICH THE GREAT NORTHERN WAS RUN.

S.S. NORTHERN PACIFIC. DESIGN { LENGTH PP'S. 500FT.0" DRAFT EVEN KEEL 20FT 3½" / BEAM MOLDED 63FT.0". DISPLACEMENT 9612 TONS

TABLE II.

STANDARDIZATION RUNS OVER GOVERNMENT NAUTICAL MILE COURSE AT RO

ENGINE ROOM DATA.

WEATHER - CLEAR AND COLD — DEC. 11 — SMOOTH SEA

		1	2	MEAN	3	4	MEAN	5	6	MEAN	7	8	MEAN	9	10
RUN NO.															
TIDE		WITH	AGAINST		WITH	AGAINST		WITH	AGAINST		WITH	AGAINST		WITH	AGAINST
TIME BEGINNING RUN		1.45 PM	2.15 PM		3.25 PM	3.43 PM		4.59 PM	5.18 PM		6.39 AM	7.08 AM		7.35 AM	7.56 AM
SPEED IN KNOTS		14.351	14.05	14.181	14.444	14.458	14.451	18.858	18.819	18.838	19.459	19.328	19.394	22.08	22.07
R.P.M. FOR RUN	P.	200.6	200.3		234.8	236.0		277.7	279.3		275.8	275.6		311.5	313.6
DO.	C.	193.4	197		223.8	223.1		263.3	268.4		261.8	261.5		295.5	293.4
DO.	S.	202.9	200		233.4	235.0		272.3	275.3		272.1	274.3		309.3	309.2
DO. MEAN		197.7	197.1	197.4	230.7	230.7	230.7	271.1	275.7	272.4	270.0	270.8	270.4	305.3	305.4
SLIP %				12.8			12.9			16.			16.1		
I.H.P. FOR RUN	P.	1230	1185		3026	2080		3600	3670		3240	3370		5100	4180
DO.	C.	1368	1400		2215	2184		3670	3740		3270	3370		5010	4980
DO.	S.	1221	1217		2005	1921		3300	3410		3280	3440		5050	4920
DO. TOTAL		3819	3802	3810	6246	6185	6215	10570	10820	10695	10490	10580	10535	15160	15020
MAIN STEAM LINE	P.	205	203	204	209	210	209	205	202	203	194	205	200	204	197
DO.	S.	204	200	202	207	208	208	202	201	202	194	203	200	200	196
MAIN STEAM ENGINE ROOM		201	199	200	203	205	205	200	198	199	188	200	194	198	190
MAIN H.P. AHEAD		126	124	125	43	46	45	210	210	210	180	180	180	178	172
MAIN L.P. AHEAD	P.	-6"	-5"	-5.5"	0	0	0	4	4.5	4.25	4	4	4	10	10
DO.	S.	-5"	-5"	-5"	1	1	1	8	8	8	9	10	9.5	15	15
VACUUM MAIN COND'N SR	P.	28.8	28.8	28.8	28.8	29	28.85	29	29	29	28.5	28.4	28.45	29.2	29.1
DO.	S.	29.6	29.6	29.6	29.6	29.6	29.6	29.6	29.6	29.6	29.4	29.4	29.4	29.4	29.4
AUXILIARY STEAM		200	201	201	213	211	212	205	203	204	196	210	203	203	199
DO. EXHAUST		3.5	3.5	3.5	7.5	7.5	7.5	7	7.5	7.25	2	2	2	1.5	1
STEAM TO GLANDS		1.5	.4	.95	.9	.9	.9	.8	.8	.8	.8	.8	.8	.8	1.
EXH. TO FEED HEATER	P.	8			10			9	8	8.5	5	5	5	2	4
DO.	S.	6.5			8			9	7	8.5	1.5	2	1.75	7	7
FEED PUMP DISCHARGE	P.	245			250			250	255	253	250	250	250	245	250
DO.	S.	250			245			240	240	240	240	240	240	250	250
FEED LINE		248	239	244	238	239	239	238	247	243	248	247	248	228	237
INJECTION		42			41			41	41	41	42	42	42	42	42
OVERBOARD DISCHARGE	P.	64			62			64	64	64	66	68	67	68	69
DO.	S.	69			68			68	65	66.5	70	72	73	69	72
AIR PUMP DISCHARGE	P.	46			50			49	48	48.5	49	48.5	48.7	50.5	50.5
DO.	S.	48			50			50	47	48.5	55	52	52.5	51	51
FEED TO HEATER	P.	80			80			80	78	79	74	78	76	70	76
DO.	S.	78			70			70	70	70	80	83	81.5	76	76
FEED FROM HEATER	P.	225			220			220	216	218	208	216	212	206	210
DO.	S.	175			140			192	210	201	210	216	213	202	198
WORKING PLATFORM		88	88		86	82		84	80	82	82	82	82	80	80
R.P.M. CENTRIFUGAL PUMP	P.	120			118			114	124	119	104	104	104	120	124
DO.	S.	114			114			110	110	110	92	92	92	112	112
AIR PUMP	P.	10			16			18	20	19	12	24	18	24.5	14
DO.	S.	13.5			13.5			20	18	19	9	9	9	13	13
FEED PUMP	O.P.	—			3			3.25	4	3.67	3	3.5	3.25	6	6
DO.	I. P.	14			13			13.5	18	15.7	10	8	9	12	12.5
DO.	I. S.	—			—			—	—	—	19	15	17	10	10
DO.	O. S.	13			13		—	13.5	14	13.8	4	4	4	13	15
BILGE & BALLAST PUMP	P.	54			40			36	32	34	52	54	53	56	48
DO.	S.	—			—			—	—		—	—	—	—	—
LUBRICATING OIL PUMP	P.	4			4			3			11	4	7.5	5.5	7
DO.	S.	9			10			3			11.5	8	9.5	11	11
DISCH. LUB. OIL PUMP	P.	40			40			40			40	38	39	40	40
DO.	S.	40			40			39			40	37	38.5	38	40
LUBRICATING OIL SYSTEM		10.4	9.4	9.9	9.9	9.5	9.7	9.8	9.9	9.85	10.1	8.2	9.2	10.1	8.1
OIL THRUST BEARING	P.	11			10.5			11			10	8.5	9.3	9	10
DO.	C.	9			8.3			8.5			8	6.5	7.3	7.5	7
DO.	S.	11			10			10			10	8.5	9.3	8.4	9
OIL FORD MAIN BEARING	P.	11			11			10.5			10.5	8.0	9.3	9.5	10
DO.	C.	11.5			10.5			10			11	8.5	9.8	9.5	9
DO.	S.	11			10			10			10	8.5	9.5	9	9
OIL AFTER " "	P.	10			9			9			8.5	9	8.8	7.5	10
OIL AFTER " "	C.	11.5			10.5			11			10	10	10	10	12
DO.	S.	8			7			7.5			7	6	6.5	6	8
OIL DISCH. THRUST BEARING	P.	78			82			86			68	77	72.5	82	86
DO.	C.	72			78			78			60	68	64	74	77
DO.	S.	78			84			88			78	78	78	88	88
" " FORD MAIN "	P.	85			89			95			80	88	84	93	97
DO.	C.	78			81			86			75	79	77	84	85
DO.	S.	84			88			92			80	86	83	90	94
" " AFTER MAIN "	P.	86			88			93			80	87	84	90	97
DO.	C.	84			84			87			73	78	77	84	85
DO.	S.	89			88			92			82	88	85	92	95
LUB. OIL DISCH. FROM COOLER		68			68			72			69	70	69.5	70	73

BOILER ROOM DATA.

	4-F. 6-AFT	4-F. 6-A.	4-F. 6-A.	6-F. 6-A.	6-F. 6-A.
NO. OF BOILERS IN USE					
BOILER PRESS. AVERAGE	209	209	209	197	210
NO. & SIZE OF BURNERS TOTAL	32 - 1 3/8" TIP	48	70	60	68
R.P.M. BLOWERS & BLOWERS IN USE	900 - 4 BLOWERS	1173 - 4 BLOWERS	1300 - 4 BLOWERS	960 - 6 BLOWERS	1000 - 6 BLOWERS
AIR PRESS. BOILER ROOM & ATTRS.	.6"	.48"	1.26"	.95"	1.05"
DO. UPTAKE	1.3"	1.64"	1.9"	1.4"	1.7"
PRESSURE OIL TO BURNERS	162	165	158	143	160
TEMPERATURE DO.	50	50	50	186	158
NO. & D.S. OF FUEL OIL PUMPS	8-EACH. 2-PUMPS	8-EACH. 2-PUMPS	10.5 EACH 2-PUMPS	8.5 EACH. 2-PUMPS	9.5 EACH. 2-P

DESIGN
LENGTH, PPS. 500 FT - 0"
BEAM MOLDED 63 FT - 0"
DRAFT EVEN KEEL 20 FT 5 1/2"
DISPLACEMENT 9612 TONS

PARSONS TURBINES
3-SHAFTS. 1-H.P. ON C.L.
1-L.P. AHEAD & 1-ASTERN
ON EACH WING.

MOSHER BOILERS
12 LARGE TUBE
TOTAL H.S. ABOUT
60 000 SQ. FT.

PROPELLERS: 3-SOLID
STR'BD. & CENTER, RIGH
DIA., 9 FT. 2" PITCH, 8 FT.
EXPANDED AREA, 38.24 S
PROJECTED " 34.05

SE AT ROCKLAND, MAINE. DEC. 11, 1914 AND
TA. DEC. 12, 1914.

SMOOTH SEA

9	10	MEAN	11	12	MEAN	13	14	15	MEAN
WITH 7.35 AM	AGAINST 7.51 AM		WITH 8.10 AM	AGAINST 8.25 AM		WITH 8.45 AM	AGAINST 9. AM	WITH 9.16 AM	
22.416	20.157	21.286	24.00	22.388	23.194	24.782	24.258	24.725	23.385
311.3	310.6		362.7	362.9		367.3	370.9	371.4	
295.3	295.6		326.0	327.1		331.0	341.4	333.9	
309.3	309.2		348.4	347.4		360.6	365.5	364.6	
305.3	305.4	305.4	341.7	342.5	342.1	353.3	359.2	356.7	357.6
		16.2			17.7				18.6
3700	3780		7420	7470		8770	9130	8950	
3010	2980		6470	6580		7700	7920	7530	
3089	2930		7220	7290		8390	8390	8210	
13760	13690	13090	21410	21140	2073	24859	25390	25270	25545
204	197	201	198	205	202	207	210	204	208
200	196	198	198	205	201	205	209	200	205
198	190	194	190	197	194	198	200	194	198
178	173	173	160	165	162	178	180	175	178
10	10	10	22	22	22	231	28	27	27.0
15	15	15	26	27	26.5	30	32	32	31.5
29.2	29.1	29.15	29.1	29	29.05	29	29	29	29
29.4	29.4	29.4	29.3	29.3	29.3	29.2	29.2	29.2	29.2
203	199	201	207	206	204	209	211	205	209
1.5	1	1.25	3	2	2.5	2.5	4	5	3.5
.8	1.	.9	1.7	2.8	2	3.2	3.8	5.5	3.5
2	4	3	2	2	2.4	4	3	5	3.3
1	1	1	1	1	1	1	1	1	1
245	250	248	245	245	245	240	250	270	253
230	220	220	250	240	245	240	250		245
228	231	230	231	227	229	239	250	248	247
41	41	41	41	41	41	41	41	41	41
68	69	68.5	69	69	69	70	70	70.5	70.1
69	72	71	68	68	68	69	70	70	69.7
50.5	50.5	50.5	52	51	51.5	52	52.3	52.3	52.5
47	53	51	53	53	53	53	53	53	53.3
70	76	73	73	70.5	71.7	74	72	73	73.2
76	76	76	77	74	75.5	76	78		77
206	210	208	204	218	211	198	200	210	203
202	174	188	186	160	173	166	178	184	169
80	80	80	79	79	79	82	78		80
130	124	127	139	130	135	143	144	148	146
112	107	110	140	140	140	140	144	142	142
22.3	18	20.3	20	20	20	21	20.5	20	20.5
13	13	13	13	13	13	13	13	13	13
6	6	6	5.5	7.5	6.5	8	14	11	12
12	12.5	12.3	14	16	15	19.5	12	4	12
10	10	10	10	10	10	13	12	—	12.5
13	13	13	13	13	13	18	16		17
36	48	42	46	37	41.5	48	32	44	42.7
—	—	—	—	—	—	—	—	—	—
5.5	7	6.3	8	7	7.5	8	8		8
8	12	10	14	10	12	12	12		12
32	40	36	40	38	39	37	36		36.5
33	40	36.5	43	38	40.5	37	37		37
10.1	8.1	9.1	10.4	10.2	10.3	9.9	9.9	9.7	9.8
9	10	9.5	11	10.5	10.8	11	10.5		10.8
7.5	7	7.3	9	8.5	8.8	8.5	9		8.8
8.2	9	8.6	10	10	10	9	9.5		9.8
9.5	10	9.8	11	11	11	11	10		10.5
9.6	9	9.3	10.5	11	10.8	10.5	10.5		10.5
9	9	9	11	10	10.5	10	10		10
7.5	10	8.7	10.5	10	10.2	9.5	9.5		9.5
10	12	11	12	11.5	11.8	11	11		11
8	9	8	9	8	8.5	8	7.5		7.8
82	86	84	83	92	87	94	93		94
74	77	75.5	83	85	84	87	88		88
84	88	86	92	94	93	97	98		98
93	97	95	98	100	99	101	102		102
82	85	84.5	90	90	90	93	93		93
90	94	92	98	98	98	100	102		101
90	97	94	100	100	100	100	103		103
84	83	84.5	89	90	90	91	90		91
94	96	95	98	99	99	102	102		102
70	75	72.5	78	79	78.3	80	80		80

TA.

		6-F.	6-A.	6-F.	6-A.	6-F.	6-A.	
		210		213		218		
		66		80		96		
		1800- 6 BLOWERS		1136- 6 BLOWERS		1309- 6 BLOWERS		
		1.05"		1.48"		1.64"		
		1.71"		2.42"		2.34"		
		160		173		170		
		158		170		176		
9	9.5 EACH.	2-PUMPS	12.5 EACH.	2-PUMPS	12 EACH.	2-PUMPS.		

S: 3-SOLID MANGANESE BRONZE.
NTER, RIGHT HAND. PORT, LEFT HAND.
PITCH, 8 FT. 4.15"
REA. 38.24 SQ. FT.
" 34.05 "

S.S. GREAT NORTHERN.

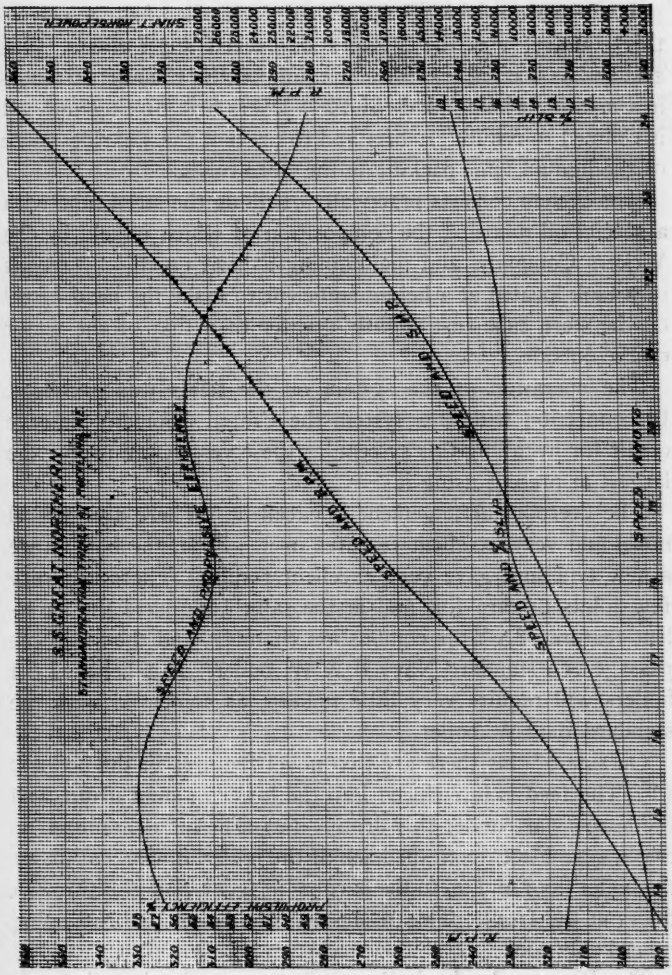


PLATE I.

low level a large 12-inch gong sounds. This prevents, by any chance, the oil level getting low. The oil is returned to the tank by a pump automatically controlled by a pressure governor of the "Ideal" make.

In the stern-tube lubrication there is another feature new to this country, viz: the Vickers oil-packed stern tube. This system has been in use for some time on the Channel steamers in England, it being made by the Vickers firm of England. It is claimed to reduce friction and to prevent the wearing down of the lignum vitae packing in the stern tubes.

The system consists practically of filling the stern tube with a special oil mixture, the composition of which is held by the Vickers Company, and maintaining a head on this by means of a tank located at a height sufficient to make the head of oil on the stern tube just slightly greater than that of the water on the outside.

The operation of the system on the trials was excellent. In connection with this system the Cramp Company deserve credit for installing this system, as these vessels are by far the largest at present using it.

The fireroom arrangement is excellent. There are several novel and very excellent features in it. First, the boilers are a new type and gave very excellent results. There are very few joints to care for, and at the same time have the excellent feature of a large combustion chamber, and the same gas passage as a standard Navy B. & W. boiler. It is believed the boilers will require very little upkeep, as compared with the standard B. & W. boiler.

Outside of this there are two novel features in the boilers: (1) The feed is introduced into the lower or mud drum, and the burners are under this drum. (2) The air is admitted in two ways: First, from the back of the boiler, down, and under the boiler floor and up to the burners in front, and also through doors at the front. This resulted in the boiler castings being remarkably cool.

The Pioneer Smoke Indicator System is also installed. It operated excellently, and gives the density of the gas and also an approximate CO_2 analysis, thus enabling the fireman to maintain an excellent combustion, and also to control the smoke nuisance. This feature is also new, this being the first

installation of its kind, and is somewhat of an experiment, as the device is not yet on the market.

The vessels are the finest merchant vessels ever built in this country and are in many ways far ahead of anything foreign builders can boast of. They are a great credit to the builders and a valuable addition to the American merchant service.

SUBMARINES AND TORPEDOES.

By Lieutenant (J. G.) C. N. Hinkamp, U. S. Navy, Member.

Interest in the submarine has reached a point hardly dreamed of five years ago. This has been brought about by the prominent part played by the submarine in the present war in Europe and by the recent catastrophe involving one of the submarines of the United States Navy.

Reviewing briefly the history of the development of submarines, the first attempt at submersible devices is found in the early diving bells. Records show these to have been used long before the Christian era. As far as is known, the first successes occurred about 1620. The first submarine was a vessel built of wood and covered with leather. Since that date, 1620, governments and individuals have been developing submarines.

In 1773 Bushnell began the construction of a submarine. It was of copper and made of two shells. The features in his boat were the arrangement of torpedoes, lead keel, ballast tanks and pumps for freeing these tanks.

The French began early to experiment with submarines, but until about 1797 very little success was obtained by them. Robert Fulton, an American, entered the field in 1800, and in 1801 began to get some satisfactory results with an experimental boat operating in the river Seine.

In 1850 William Bauer, a German, drew plans for a boat, and in 1856 built one for Russia. This boat was propelled by treadmills. It was lost, however, and no more of the type were built.

The first period in which submarines were actually employed in naval service was during the American Civil War, and no

really successful operation of this type of vessel is found until that time.

In 1863, a submarine designed by Alstill was built in Mobile, Alabama. This boat was not used, but had several features in its design which are still considered excellent.

The next series of boats were classed as "Davids." They ran partially exposed and could hardly be called submarines. The Davids were more or less successful; some were lost, but their successes in the Civil War paved the way to the weapon of the present day.

In 1863 the French again took up the problem. Their vessels were designed to move along the bottom, but very little actual experimenting was done with them. Their success was indifferent.

In 1881 the Nordenfeldts took up the design of submarines, and in 1885 built a vessel for the Turkish Government. The propulsive power for this boat was steam.

In 1888 the Goubet submarines appeared, with indifferent success.

The next design appeared in 1897 and was the creation of M. Laubeuf. This type was really the first example of the modern submarine, so it may be said that the date of modern submarine construction is 1897, or just prior to the Spanish-American War.

The inventors in the United States were not idle during this time. The two famous American inventors, Holland and Lake, were each perfecting a type of boat and these types have now been adopted as practically the standards of the world. Holland's experiments began in 1875, but it was not until 1893 that his plans were approved by the American Naval authorities. In 1904 Lake's boats were built for Russia.

Up to 1898 the submarine was in the embryonic state. From then up to the present time it has developed rapidly and great strides have been made in its design with advances in the military features as well as sea-keeping qualities.

In the United States the development has been rapid. The present appropriation for the building of submarines calls for

two distinct types, sea-going, of 1,200 tons, and coast or harbor-defense submarines, of about 400 tons. The contracts for these have recently been awarded. These types present added features making for increased comfort for the crew and embody many features to increase the military efficiency.

In the construction of a submarine, consideration has to be given to two kinds of tanks, namely, ballast tanks and trimming tanks. The ballast tanks are divided into main ballast tanks and auxiliary ballast tanks. The trimming tanks are tanks in the bow and stern of the boat used for trimming the vessel in the fore-and-aft line. The ballast tanks destroy the maximum part of the reserve buoyancy when completely filled. The smaller ballast tanks are used finally to trim the vessel to a predetermined amount of buoyancy. In addition to the ballast and trimming tanks the vessel is fitted with fuel tanks to carry the necessary fuel for the main engines.

The torpedo tubes are located in the bow of the submarine. In some classes there are surface tubes and tubes in the stern, but ordinarily they are fitted only in the bow. In any case, it is necessary to aim the boat whenever a torpedo is to be fired.

The engines for propelling the vessel on the surface are located in the stern as are also the main motors.

The storage batteries for submerged navigation are generally located amidships.

In the following an attempt is made to describe the use of this combination of tanks, engines, batteries and torpedoes.

It should be understood that there is nothing mysterious in the operation of a submarine. The orders used in the handling of a boat are few; they are made as comprehensive as possible and are so given as to eliminate any possible confusion.

Preparing to submerge includes all preliminary work up to the closing of the conning-tower hatch. This comprises the stowing of the deck gear, taking down the bridge, unrigging the radio, closing the hatches, unlocking the valve-operating mechanism, securing the engines; in fact, a clearing ship for action. This operation requires from two to twenty minutes, depending on the amount of rigging to be taken down.

Before proceeding with the description of the actual submerging of a boat, a few definitions of terms used will be given:

Positive Buoyancy.—That condition in which a boat any part of the structure of which is above water requires ballast to sink.

Negative Buoyancy.—That condition in which a submerged boat without headway requires the removal of ballast to rise.

Neutral State.—That condition in which a completely submerged vessel is in equilibrium.

Submerged Trim.—The amount of positive or negative buoyancy a vessel possesses with reference to the neutral state.

Awash Condition.—That condition in which the main ballast tanks are empty and the auxiliary ballast and trimming tanks contain the requisite ballast so that by flooding the main ballast tanks the vessel will have the proper trim for submerged operations.

Flooding.—That process whereby water is admitted to a tank directly from the sea.

Filling.—That process whereby water is pumped into a tank.

Blowing.—That process whereby water is forced out of a tank by means of air pressure.

Pumping.—Pumping water overboard from the tanks by means of main or auxiliary pumps.

The actual submerging of the boat can be done in two ways, one called the "static" dive, the other the "running dive."

In the static dive, also known as "balancing," the boat is submerged but does not move except in the vertical plane. This dive may be accomplished in two ways: by trimming the boat and maintaining her trim by adjusting the ballast, or by dropping the anchor, trimming the boat to within a few hundred pounds positive buoyancy, and the heaving in or veering on the anchor cable. The latter way is the simpler method for easy control and can be used where there is no current, or only a small amount of current, if the sea is not too rough.

The principles involved in balancing a submarine are merely those by which is determined the force of gravitation. The

mass of the submarine itself amounts to several hundred tons, but the actual forces used to sink it from a neutral state are very small. The addition of 50 pounds of water will cause the boat slowly to descend. As soon as the vessel has started on its downward path it is necessary to remove part of the ballast in order to keep it from continuing on this path, provided the density of the water is constant. By pumping out, or blowing, a given quantity of ballast, there will be a force exerting an upward tendency. The determination of the amount of ballast necessary, the forces, is the difficult part of balancing. In balancing by the use of the anchor, the different forces do not have to be considered to such a great extent.

Before submerging, the vessel is generally brought to a fore-and-aft trim which will cause the boat to be level when submerged. This is done by flooding or filling the forward or after trimming tanks.

Balancing by means of the anchor is, roughly, accomplished as follows: The anchor, which weighs about 1,000 pounds, sometimes more, is dropped. The boat is then trimmed down until it is in practically a neutral state, or has an amount of buoyancy equivalent to two to three hundred pounds. It can readily be seen that with a reserve buoyancy of three hundred pounds and a weight of a thousand pounds on the bottom there is a preponderance in favor of the anchor of about six hundred to seven hundred pounds. By slowly heaving in on the anchor cable, the positive buoyancy of the boat is easily overcome and she may be drawn down to any depth desired.

The static dive by adjustment of ballast is made as follows: After getting the fore-and-aft trim, as already described, the main ballast tanks are flooded. These tanks are flooded through large kingston valves, the operation requiring from one to two minutes. There is an enormous volume of air in the ballast tanks that must be freed before the tanks can fill with water, and the process of getting rid of this air is called "venting." The main ballast tanks "vent" overboard, as the large volume of air, if admitted into the interior of the boat, would cause the barometric pressure in the boat to rise to such

a point as to be uncomfortable. All the other tanks, being comparatively small, vent inboard.

The boat being trimmed down as far as the main deck, still has too much buoyancy to run submerged. The tank next flooded is the auxiliary ballast tank. This tank holds enough water to destroy the remaining buoyancy and is flooded generally until the top of the conning tower is just under the surface of the water. The final trimming is done by slowly flooding or filling the adjusting tank. When the vessel is trimmed until there is about two to three hundred pounds of positive buoyancy, it can be readily handled submerged. This is considered the best trim for all around work, and completes the static dive. From this condition any operation submerged can be commenced.

The state of the sea affects balancing. In trimming, the boat oscillates until it strikes a condition of comparative equilibrium, when it comes to rest. With a "sea" on, this state of equilibrium is never reached, and the amount of water taken in to approach a neutral state is difficult to estimate owing to the fact that the amount of positive buoyancy depends on the area of the water-line plane which varies with every wave that passes. Suppose a vessel rising to the effect of a passing wave adds a little water in the ballast tank. As soon as the wave passes, the boat has a tendency to sink and, with the added impetus caused by additional ballast, it falls rapidly. If the "downward send" is sufficient, the only thing that will stop the vessel is the bottom. The downward velocity may be even great enough to overcome a fairly large amount of buoyancy.

To be able to determine when to stop is almost the entire secret of the art of balancing. For strategical purposes it is better to have a small amount of headway on the boat, just enough to overcome the effect of the sea which, being rough, would hide the wake of the periscope in the foam of the white caps.

The running dive is made from the awash condition. In the awash condition the trimming tanks and auxiliary ballast tanks are flooded to the amount necessary for the proper trim

when submerged; the main ballast tanks are empty. The running dive is used for all tactical purposes except balancing.

The vessel being underway "awash," the order is given to submerge. All hands get into the boat, the engines are stopped and the electric motors started. As soon as the engines are stopped the conning tower is closed, all ventilators housed and the main ballast tank flooded. Knowledge that the trim will be approximately correct when totally submerged renders careful adjusting of ballast unnecessary. The boat is inclined slightly, about one-half of a degree down by the head, and the inrush of the water controlled by manipulation of the valves. All this is done in the short period of from one to two minutes. Seven boats have been known to average about two minutes for this evolution during maneuvers. If the control of the boat is difficult, ballast is taken on until the boat "handles" correctly; then trimming is stopped.

The duties of each member of the crew are clearly defined. At least two and sometimes all the men are thoroughly familiar with each station, so that when an order is given it is intelligently and promptly carried out. All orders are repeated. When an order has been executed the fact is reported to the commanding officer. In submerging a boat all precautions are observed before flooding the main tanks, and the fact that they have been followed is reported to the commanding officer or his assistant by the men at the various stations.

Submerging a submarine is distinctly a one-man job. The commanding officer must be thoroughly conversant with all the details of the actual submerging of the boat and he must at all times be thoroughly informed as to existing conditions in the boat. None of the important features can be delegated to anyone else, as each condition or state of affairs has a distinct relation to every other condition. The second officer assigned to the submarine generally assists the commanding officer where possible, and endeavors to acquire all the information possible on the details of the operation in order that he may take charge in case of any emergency.

Navigating a submarine on the surface is just like navigat-

ing any ship of the same size except that the displacement for amount exposed is greater. Care must be taken in making landings, as the headway is not so easily checked due to the available power for this purpose being less per ton generally than in ordinary vessels. Navigating submerged, that is, totally submerged, is just like navigating a ship in a fog on the surface, except that when the depth is great enough there are no passing steamers and fishing craft with sleepy crews to cause worry. Navigation submerged may be compared with the flight of an aeroplane in all respects except speed and the density of the substance through which passage is made. There are, of course, no "holes" in the water as there are in the air, but there are cross currents, strata of different density and other conditions to consider. A submarine uses the diving rudders for submerging or rising in much the same manner that a hydroplane or aeroplane uses the horizontal rudders to obtain the necessary angles for rising or gliding.

Those who have observed an aeroplane in flight have, no doubt, noticed the rolling of the machine. This is apparent in a puffy wind. In a submarine running submerged we have the same condition if the water on the surface is rough. A submarine rolls a slight amount at a depth of 50 feet if the surface is *very* rough, but practically all motion is lost at a depth of 75 feet. During the recent maneuvers in the English Channel, however, commanding officers of submarines reported that it was difficult to find a depth at which there was no motion. This is readily understood, as the waters in that vicinity are somewhat shallow and the ground swells can be felt quite near the bottom.

Depth is maintained by the use of the diving rudders. Down rudder sends the stern up and the bow down. As soon as the boat is inclined downward the plane of the deck gives a large surface for the water to act on, and the power delivered to the propellers may be resolved into two components, the horizontal, driving the boat ahead, and the vertical, driving the boat downward. The amount of the vertical component of the power delivered is controlled entirely by the diving rudder.

The hull is so built that normally there is a tendency for the boat to rise, brought about by the angle of the hull and the unbalanced pressures caused by inequalities in the areas of the surfaces on the bottom of the boat and on the top of the boat. This tendency must be counteracted by the trimming of the vessel and the action of the diving rudders. In some designs of submarines there are hydroplanes, forward diving rudders, and other devices to perform the same work, the difference being in diving at an angle or on an even keel.

The steering of the vessel in azimuth is done the same submerged as on the surface, using either hand or electric power. The magnetic compasses for submerged running are mounted above the hull in a composition hood and so arranged that the face of the card is reflected into the hull where the helmsman can readily see it. In recent years the gyroscopic compass has been perfected and has been installed in most of our submarines. This compass is operated entirely by electricity and the force of gravitation, both of which can be made independent of the magnetic influences of the earth. The gyroscopic compass must be adjusted for the different speeds and courses, but except for this it is automatic in its action and points in a true meridian with very little or no variation. The installation of gyroscopic compasses renders the tactical problems far simpler, as it is possible with this compass to steer the vessel within narrower limits than was possible with the magnetic compasses, owing to the fact that the action of the gyroscope is instantaneous, while the magnetic compass is very sluggish.

Control of the submarine is affected by speed. The effect of rudder manipulation varies in direct proportion to the speed of the boat. It is possible to handle a badly trimmed boat if sufficient speed is used.

It has been demonstrated that there are quite marked differences in density of water at different points and that this variation in density has a very appreciable effect on the trim and handling of a submarine. An account of an actual experience of certain United States submarines in which variation of density of water necessitated a difference of as much

as 150 to 200 pounds of ballast per mile will be found in an article entitled Notes on Submarine Repairs and Operations by the author of this paper in the JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS for August, 1914.

The periscopes are the eyes of the submarine. There are many styles of periscopes—monocular, binocular, stationary, revolving, rotating eye piece, stationary eye piece and "walk around." There are also devices known as omniscopes in which the entire horizon is reflected on a ground glass and any part can be separated from the rest by means of screen sectors. The periscopes are long tubes with lenses in the top and bottom and so fitted with prisms that the rays of light are paralleled. The general impression one gets is that of looking through a telescope.

The periscope that is in most general use in the United States Navy is the walk-around type with normal and magnifying eye pieces. Each boat is required to have at least two periscopes, one for the commanding officer and one for the helmsman or the second officer. In the recent boats, fitted with gyroscopic compasses, the helmsman takes his station at the compass while the lookout is at the periscope from which he directs the helmsman. The commanding officer usually stations himself at the after periscope and directs the movements of the vessel. Should he leave the periscope for any purpose, such as to consult chart or to get signals, some one always takes his place, so that there is never a time that both periscopes are not manned. When totally submerged, however, no one is needed at the periscopes for it is impossible to see more than a few feet through the water.

Signaling while submerged is a subject of much interest. In the early days of submarine navigation signaling under water was done in a most crude manner for which the hull of a vessel was found to be peculiarly adapted. Sending was accomplished by tapping on a rivet with a hammer and receiving by holding the forehead to a frame of the boat. For several years inventors have investigated and experimented extensively with the result that there are now in practical use in

submarine signaling the submarine bell, the Fessenden oscillator and the vibrating wire, by which it is possible to signal effectively at distances greater than five miles under favorable conditions. All these systems set up vibrations in the water which are detected by microphones and heard through the ordinary telephone receiver.

Inventors are now endeavoring to devise means for increasing the speed of transmission.

Each submarine is equipped with one bell, some with two, and on every submerged run, in peaceful maneuvers, these bells are rung so that it is possible for those in one boat to know the approximate location of other boats. The fact that no signals were received from the *F-4* has caused a belief that the vessel was so damaged that the crew remained alive for but a very short time after the accident which sank her occurred.

A phenomenon is the submarine "echo." The sound waves strike an obstruction, are reflected, and are picked up at the source. This could be used to determine approximately distances from obstructions, dangers, etc., but is not recommended as the possibilities of error are many.

On the surface submarines make use of the ordinary means of signaling—wig-wag, hand-semaphore shapes, flags and radio. The last is the best, but is not developed to that degree where it is absolutely reliable in submarines except under ideal conditions. However, it has proved its worth. Shape signals must necessarily be small owing to lack of space to handle them. Flag signals are inadequate for the same reason, but are carried for use in port. At night, light and pistol signals are used. Either of these systems is effective. A means of signaling which can be used very effectively both in the day time and at night is the flashing of the searchlight. While it seems rather unusual to employ a light for signaling in the day time, it can be used very effectively, and occasions have arisen when signaling with a submarine at a distance could be carried on in no other way.

The equipment of a submarine for the comfort of officers and crew is not very extensive. Officers generally sleep on

cots and the crew in hammocks. Ice boxes are installed on the later boats, enabling them to carry fresh meats and vegetables. Meals on these vessels are, therefore, more satisfactory than on the earlier boats. In addition to the fresh provisions carried, each vessel has a dry-food supply constituting an emergency ration sufficient to last 5 days. Fresh water is carried in tanks, and it is necessary that the crew be put on an allowance when at sea and that bathing be done over the side.

The submarine of the present day can operate at its maximum speed submerged for about one hour. At about one-third of this maximum speed she can operate practically twenty-four hours.

The offensive weapon of the submarine is the torpedo, or if necessity demands, the vessel itself can be used as a ram. The effect of the submarine as a ram may be illustrated by the fact that in a collision between a United States vessel and its tender several years ago, the tender was so badly damaged as to require beaching in order to save her.

The principal offensive weapon is, however, the torpedo. Ramming would be resorted to only in an exceptional case. The torpedo is an intricate mechanism built along the lines of the submarine itself, but automatic in its action after leaving the tube from which it is fired. The torpedo is divided into four main parts—the head, air flask, afterbody and tail. The head contains the explosive and the mechanisms to fire it. These heads are carried on the torpedoes only in war time. In time of peace exercise heads containing water are used. Abaft the head is the air flask. This contains the compressed air for the propelling machinery. In the next division of the torpedo is found the depth-controlling mechanism, the air superheater, the main propelling engines, the steering engines and gyroscope, and the shafting to the propellers, as well as the rods to the rudders. The tail inside contains the gear wheels for the propellers; on the outside are the supporting parts for the propellers, the diving rudders and the vertical rudders. The early torpedoes had an effective range of about

500 yards. The most modern product has an effective range of more than 10,000 yards. The development of the torpedo has been as amazing as the development of the submarine.

The operation of a torpedo is very similar to that of a submarine, except that while the submarine is operated by the crew in it, the torpedo is operated entirely by the automatic mechanism it contains.

The primary object of a submarine is to fire torpedoes, and all other considerations must be subordinated to this as far as the tactical value of the vessel is concerned, but not to such an extent as to lose sight of the fact that the torpedoes cannot be fired at a target unless the vessel arrives on the scene of action. The efforts of the crew are aimed towards one thing, sinking an enemy, and the placing of the boat in position to accomplish this end calls for the coöperation of all the departments in the boat.

An attack on a hostile vessel can be made in so many ways, and would have to be made under so many different conditions, that it is impossible to lay down any hard and fast method of procedure. The best time to attack is at dawn or twilight, or at a time when there is enough sea running to make the detection of the presence of the boat most difficult. Submarines have been known to approach within five hundred yards of a battleship in a choppy sea and in a fog to come up under the stern before being discovered, notwithstanding the fact that lookouts were stationed on the battleships to watch for the submarines. On smooth days the wake of a periscope may be picked up at a distance not greater than 5,000 yards. A submarine cannot, however, be distinguished at much more than five miles distant, at which point the boat would usually submerge and prepare to attack.

The best situation for an attacking submarine is forward of the enemy's beam and approaching her. From such a position it is possible to get within range most easily and lie in wait until the enemy crosses the path of the submarine. In such a case it is possible for the submarine to pick the time for firing with some degree of assurance that the torpedo will reach the

enemy and destroy her. Should the enemy be very much faster that the submarine, tactics become a case of hare and hound, with the advantage in favor of the enemy, as the speed of a submarine is limited.

The subject of safety in the submarine has received much attention and many devices have been invented designed to obviate the dangers said to attend the operation of this type of vessel. Each designer and builder has his own method of escape for the crew in case of accident.

It is believed that the one most effective and most nearly positive means of safety is eternal vigilance. In reviewing all the accidents that have occurred in the last twenty years, hardly any are found that cannot be attributed to some one thing that could have been prevented had the commanding officer been able to correct the fault. Submarines should be so operated, and it can be done, that all except remote possibilities are provided for.

In the United States Navy officers and men on submarine duty are given a thorough course of instruction. The men are required to be thoroughly conversant with all the parts of the submarine before they are qualified to receive the extra compensation allowed for submerged runs. The officers under instructions are trained by experienced officers and are not allowed to operate alone until they have shown that they are capable of doing so. This system has worked most successfully, the recent accident that caused the loss of the *F-4* being the first serious accident which has befallen a submarine of our Navy. The training goes on continually and all hands are impressed with the fact that submarine operation is most important duty.

It is true that accidents can and will happen, but conditions which make for catastrophe generally do not arise so rapidly that some means cannot be taken to get the boat up and the crew to a place of comparative safety before the actual accident takes place, provided, of course, that the vessel does not get beyond her safe depth. In case of accident at a depth exceeding 100 feet, the chances of getting out are remote. Even if

a man escaped, he would have little chance of getting to the surface alive owing to the sudden changes of pressure.

The mental condition of officers and crew in the event of accident should receive most careful consideration in the design of any means for escape, and the aim should be to produce a device that can be operated with certainty by even the panic-stricken.

As submarine design improves, however, dangers of operation are becoming fewer. Today the submarine possesses no more inherently dangerous features than are possessed by any ship. In fact, they are safer to cruise in than a surface craft and for the same tonnage are far more comfortable in a heavy sea.

The personnel of a submarine depend for air for breathing purposes while submerged on the free air in the boat at the time of submerging and the compressed air carried in storage flasks, which is used in freeing ballast tanks of water as well as for breathing. In the average submarine in commission today, the air contained in the boat at the time of submerging is sufficient to last officers and crew, numbering 18 men, for a period of from 9 to 12 hours. The air carried in the storage flasks is about sufficient to replenish the entire volume of air in the boat twice at atmospheric pressure, provided it is used for no other purpose. The maximum time during which all the air available can be breathed without serious effect is, therefore, from 30 to 36 hours. In computing this time, the safe amount of CO_2 that should be allowed to accumulate in the air at any time is taken at 2 per cent. Men vary, however, in their ability to withstand the effects of CO_2, the average man being able to withstand about 2½ per cent., while an exceptionally strong man can withstand as much as 5 per cent. Therefore, the time during which life can be sustained by the air in a given boat will depend somewhat upon the powers of resistance to the effects of CO_2 on the part of the personnel.

The air is maintained in condition for breathing by two methods: first, by slowing bleeding from the main air supply into the boat and pumping air out very slowly; and, second,

by allowing the air to become foul and then pumping a part of it out of the boat and replenishing it from the air flasks, at the same time maintaining a normal atmospheric pressure in the boat. The former method has been found more economical of air. The air in the boat is kept in circulation, as it has been demonstrated by experiment that air in circulation may be charged with a much greater percentage of CO_2 than still air without evil effect on the person breathing it.

THE FUTURE OF THE SUBMARINE.

The future of the submarine opens a large field for speculation. It is believed that the size of the boats, the general design, the motive power and the principal characteristics will not change materially in the next few years, except in the case of the sea-going submarine which is now being built and which departs greatly from all previous designs.

As far as can now be foreseen, the future power of submarines may be expected to be the oil engine or some modification of it. However, engineering fields have not been sounded to their depths in the determination of adequate power for these vessels, and it is believed that the future may see some development and improvement, and perhaps a material departure from any methods of propulsion heretofore adopted or proposed.

With several flotillas of the sea-going type of submarines above to cruise with the fleet, they might one day be found in the line of battle with the destroyers. Some authorities claim that this type of submarine will eventually supplant the torpedo-boat destroyer. This is considered most improbable, however, as, owing to their shape, high speeds call for prohibitive powers, and, with the methods of propulsion now in use or contemplated, it is not reasonable to expect a speed greater than 25 knots on the surface or 15 knots submerged.

NOTES.

THE BATTLE-CRUISER IN WAR.

By Archibald Hurd.

(*Author of "Command of the Sea," "Naval Efficiency," etc.*)

Every notable success of the British Navy in the present war has been due to the battle cruiser. By the employment of ships of this new type the enemy on three successive occasions has been surprised owing to the superior mobility of the British force, and then overwhelmed by superior weight of metal in combination with, at least, not inferior gunnery.

The triumph of the battle cruiser bears testimony not only to the foresight exhibited by the British Admiralty, but attests the competency with which they applied the historical principles of war to the new conditions which were already coming into view when the battle-cruiser type was evolved. Had the British Fleet not possessed ships of this class, the discomfiture of the enemy in the Bight of Heligoland in August last would have been impossible; Admiral Graf von Spee's squadron could not have been practically annihilated in the course of a few hours on December 8th; and on January 24th, when the enemy set out to bombard once more the East Coast of England, his retreat could have not been cut off, the *Blucher* sunk, and two other large units seriously damaged. But for the battle cruisers, the British Navy could have made little progress during the past seven months in weakening the enemy, and would not have had much to show as an offset against our losses inflicted by the Germans by the use of submarine and mine. Generally it may be said that the Germans owe practically everything to invisible attack, and the British to swift surprise, supported by heavy gun power. Commenting upon the Dogger Bank action, the First Lord of the Admiralty in his speech in the House of Commons, on February 15th, remarked that it vindicated, "so far as it went," theories of design, and particularly of big-gun armament, always associated with Lord Fisher. The range of the British guns was found to exceed that of the Germans. Although the German shell is a most formidable instrument of destruction, the bursting, smashing power of the heavier British projectile is decidedly greater and—this the great thing—our shooting is at least as good as theirs.

What manner of ship is the battle cruiser which has contributed so greatly to the promotion of the British cause on the sea. Mr. Kurt Orbanowski, a naval constructor of Polish birth, has recently given an interesting definition of this type of man-of-war. Mr. Orbanowski was formerly with the Hamburg Shipbuilding Yard, and was responsible for the design and construction of the armored cruisers *Scharnhorst* and *Gneisenau*, and, later on, of the battle cruisers, *Moltke* and *Von der Tann;* subsequently when the Russians established their large shipyard on the Gulf of Finland, he was placed in charge of it, and designed the battle cruisers of the *Borodino* class of 32,000 tons, a nominal speed of 27 knots, an armored belt of 13 inches, and an armament of twelve 14-inch guns in association with twenty-one 5.1-inch guns.

Dealing with the battle-cruiser type, Mr. Orbanowski recently declared that without a sufficient number of this type a battle fleet remains a torso,

apt to be attacked at a tactically disadvantageous position by a powerful adequately equipped enemy. "A battle-cruiser squadron as a fast division, or fast-division wing, of the battle line, can, because of superior speed, quickly accumulate power at the enemy's weakest point, attacking head or rear unexpectedly, or repulsing similar attacks of the opponents. Their strategical value is without doubt. They can cut off and repulse the enemy's scouting forces (cruisers) and throw a complete veil over the movements of their own battle fleet, especially on the high seas, where the use of air craft is still very limited." Their speed, he contended, enables them to rush to points of a coast menaced by an enemy's raids, which is especially valuable for a country with such far-extended and double-sided coasts as the United States, with many great industrial towns in reach of attack. "Their powerful armament holds the enemy in check until battleships arrive. The essential qualities of a battle cruiser are heavy guns of the same, or nearly the same, caliber as those of battleships, slightly less in number; slightly less thickness of armor; and high speed." According to the investigations of naval experts with whom he has been in touch, Mr. Orbanowski stated, their speed should be twenty-five or thirty per cent.—or at least 5 knots—more than that of the battleships. As battleship speed, up to the latest types, was 21 knots, battle cruisers made 26 to 28 knots. The increased speed of new battleships to 22 or 23 knots necessitated an increased speed of 29 to 30 knots for battle cruisers. "Higher speeds than these at the present stage of marine engineering, aside from any unexpected developments, cannot, in his opinion, be counted upon.* "Therefore we meet already for solution the problem of the intermediate type of battleship, using oil fuel only, such as the British *Queen Elizabeth*,† or the Italian type, with 25 knots speed."

These considerations led the two leading naval Powers now engaged in war, Britain and Germany, to develop the battle cruiser side by side with the battleship of the Dreadnaught type. Japan, and much later, Russia, followed the example. Smaller nations, such as Italy and Argentine, adopted the intermediate-type of speedy battleship (23 to 25 knots). The British battle cruisers number about thirty per cent. of the number of the modern battleships, even if we take into account the intermediates of the Elizabeth type (25 knots). There are 4 Invincibles, 4 Lions, 2 New Zealand and Australia, altogether 10; against 23 Dreadnaughts (21-23 knots) and 5 intermediate Dreadnaughts (25 knots)—altogether 28; plus 1 Queen Elizabeth (27,000 tons) and Royal Sovereigns (25,300 tons) now building.

Germany has 7 battle cruisers, 1 Von der Tann, 2 Goeben, 1 Seydlitz, 1 Derfflinger, 2 Luetzow; against 17 Dreadnaught battleships, 4 Nassaus, 4 Helgolands, 7 Koenigs, 4 Kaisers; besides 1 battle cruiser and 1 Dreadnaught building, *i. e.*, nearly forty per cent. of the number of their battleships. Russia's naval program proposes a similar proportion, *i. e.*, battle cruisers one-third of their battleships. She has already four big units of 33,000 tons (29 knots) on the stocks. It is a notable fact that down to the outbreak of war neither the United States, France, Austria-Hungary, or either of the South American or Latin American Powers had laid down a single battle cruiser. Why? The matter is one of some interest in view of the leading events of the present war on the seas.

The success with which the British Fleet has been able to employ these ships throws into prominence the criticism with which the new type was assailed, criticism, moreover, which apparently was regarded by a large number of British and foreign naval officers as conclusive. When the first three battle cruisers of the Invincible type were laid down and then for two years in succession no provision was made for other units of the

*In this statement Mr. Orbanowski will probably be shown to be in error.
†The British Royal Sovereign battleships are also to have liquid fuel only.

same character, an anonymous writer in the "Naval Annual" declared that it was apparent that the British Admiralty were "not going to build any more huge armored cruisers," and it was added that "we must hit upon some plan of employing those which we have already." Admiral Sir Reginald Custance was equally decisive in expressing his opinion of the future of the new type of man-of-war. He claimed that "by argument the class have been killed" and it only remained "to inter them decently away from the public gaze." Admiral Sir Cyprian Bridge, in the spring which witnessed the appearance of the first British battle cruisers, wrote an interesting article on the lessons of the war in the Far East. He declared:

"A ship of war is intended principally to fight and not to run away. We should therefore be careful not to give to any other element undue predominance over the element of offensive power in the design of a ship meant to be capable of destroying or defeating her antagonist. In ships for fighting general actions—that is ships for fighting in combination with consorts, the element of offensive power in any individual ship should bear the proper relation to the aggregate of that power in the whole group. Suitable dispersion should be given to the instruments of offensive power, and allowance should be made for suitable concentration of their effect. For certain classes of vessels, which usually will be of small size, very high speed, greater than that of an antagonist, if possible, should be provided; but it must be understood that these vessels can play only a special and restricted part in war."

It should be added that this statement was written before anything was known of the design of the Invincible class. Sir Cyprian Bridge confined himself to a statement of views based upon his interpretation of the action which had recently been fought in the Far East.

The most strenuous opponent of the new type was Sir Reginald Custance. By the time this officer wrote, a good deal was known of the new ship, and he held that he was in possession of sufficient facts to justify criticism. He contended in "Naval Policy; A Plea for the Study of War," in the first place that "neither in practice nor in theory has it ever been proved that superior speed gives any tactical advantage, unless it be thought an advantage to be able to run away.

"* * * Speed is not a weapon and does not give protection, except in running away. The aim should therefore be to endow a fleet not with superior speed or protection, but with superior offensive power, *i. e.*, gun-power." He denounced the Board of Admiralty which was responsible for the new design, claiming that it consisted of men "who had no practical war experience and, not having studied history, are not familiar with its leading principles." He added that, "Our belief is, that future wars will show the necessity for a large number of capital ships moderate in size, supported by a small percentage exceptional in power; that as these ships are to act together their speed should be the same, and should be that best suited to the more numerous class; that the exceptional ships should excel in fighting power, and not in speed."

In 1912 Sir Reginald Custance published a book entitled "The Ship of the Line in Battle," and in the course of his discussion of the co-related problems of gun-power, armor and speed, he first of all dealt with the armored cruiser and contended that "the armored-cruiser policy was not based originally on true conceptions of war." It involved, he urged, not only a faulty strategy, but mistaken ideals, both of which reacted on ship design. "The essential feature of the armored-cruiser design was a sacrifice of fighting power to mobility in the ship of the line. If it is admitted that the policy was based on faulty strategy and tactics, that sacrifice cannot have been justified. The proportion of fighting power allotted to armor in the ship of the line may have been excessive, but what justified the sacrifice of guns? What are the reasons for it now?

Are they tactical? Is it quite certain that in war games and tactical exercises during peace undue importance is not attached to supposed tactical advantages, and that actual fighting is not relegated to a secondary place in the mind? Are the conclusions based on true premises? In time of peace make-belief under mistaken ideals may be practised with impunity, but in war there is no make-belief. Then realities will have to be faced, as by the Russians at Ulsan."

Towards the end of this interesting volume, written by an officer with considerable sea experience, a student of naval warfare and a former Director of Naval Intelligence at the Admiralty, certain specific principles were laid down which in the light of events may be profitably recalled. Dealing with the battle of Shushima, Sir Reginald Custance observed that the facts "confirm previous war experience that the danger to the flotation and stability is not great. Is it worth while to divert from the guns the great weight required to give effective armor protection to the waterline, when the chances are that the battle will culminate before it is hit? Will it not suffice to make sure that the magazines are safe from direct blows and, for the rest, to trust to watertight sub-division, to armor only so far as it may limit the size of such holes as may be made, and above all to gunfire to beat down that of the enemy? Is it not more important to disarm the enemy than to sink him? Are not the protection of your own waterline and the perforation of that of the enemy secondary considerations in settling the armor and guns to be carried?"

Sir Reginald Custance then dealt in general terms with what has come to be known as the all-big gun principle:

"Now the main object in battle is to make the enemy believe that he is beaten. The most effective way to do this is to disable his *personnel* and silence his guns. The above results (that is the results of the battle of Tsushima) seem to indicate that the smaller gun is by no means to be neglected as an instrument for this purpose. The effect produced depends, not only on the size of the projectile but on the place where it hits. A small shell on the right spot is more effective than a large shell in the wrong one, but to hit the right spot is difficult. Hence, in determining the armament of a ship, a careful balance must be maintained between the numbers and sizes of the guns carried. Again, the facts show that it is misleading to compare the gun power of ships by the total weights of their respective broadsides. To do so is to assume that on the average an 850-pound 12-inch shell will damage the fighting efficiency of the ship as much as will eight 100-pound 6-inch. Such an assumption seems not to be true. When the guns in ships of the line were all about the same size, the method was legitimate, but it is believed to be entirely misleading at the present time when they differ so much, some being, perhaps, unnecessarily large and others too small for the work to be done. Are not the numbers and sizes of the guns carried the best and only safe standard of comparison?

"Thus we see that whether we consider the difficulty of hitting or the comparative effect produced by shells of different calibers, there are grave doubts whether batteries of comparatively few large guns form the most effective armaments. Any reduction in the size, or change in the disposition, of the guns will at once react on the size of the ship. All vertical armor in existing ships is now perforable by high-explosive shell, and does not seem to return sufficient value for the weight it absorbs and diverts from guns, except against fragments of bursting shells. Moreover, ships can be beaten without perforating the armor—*e. g., Orel.* The smallest gun required to maintain the ascendency over the armor must be carried in sufficient numbers, but should not these be supplemented by others of smaller size to increase the volume of fire and to provide a larger margin for misses and failures? Ascendency over the armor means that victory can be won in spite of it, that is to say, by

perforating it if spread out, and by overwhelming the ships and crew if concentrated."

No apology is necessary for quoting at considerable length the varied and instructed criticism with which the battle-cruiser type was assailed, particularly as this line of criticism was apparently endorsed by the Navy Department of the United States. The battle cruiser was denounced as a ship faulty in design, the principal characteristic of which was ability to run away and inability, owing to the absence of small guns, "to make the enemy believe that he is beaten." It was, moreover, repeatedly affirmed that such vessels were really capital ships, and as such would be massed in fleets with the battleships; no admiral, it was contended, would have the courage to spare such units for detached service.

During the course of the war no action has occurred in which the battle squadrons of either the British or German Fleets have been engaged. On the other hand, the Germans have on three occasions employed a force of battle cruisers, the objective in each case being to bombard defenceless towns on the East coast of England. Why? it may be asked, should battle cruisers have been employed, since this type of ship is little faster than a number of the light cruisers under the German flag? In the first place, the big ship holds its speed better in a heavy sea; in the second place, the big ship carries guns which can overwhelm small craft. The intention of the Germans in each instance was to cross the North Sea during the hours of darkness. If British light cruisers or destroyers were encountered, the heavy gun could put them out of action; if submarines were met with, the high speed of the battle cruisers would in itself prove a protection against attack. The first two attempts to carry out this policy were successful in that the German ships reached the British coast, fired their guns and returned home without loss.

It was on the third occasion—on January 24th—that an exhibition was provided of the value of the bigger gun carried in the British ships and of their superior speed. The Germans employed on this occasion—the *Moltke,* credited in the "Naval Annual" with a trial speed of 28.4 knots; the *Seydlitz,* reputed to have obtained 29.2 knots; the *Derfflinger,* with a rate of steaming superior, it was believed, to that of the *Seydlitz;* and the *Blucher,* a contemporary ship to the British *Invincibles,* with a trial speed of 25.3 knots. Admiral Sir David Beatty's Squadron consisted of the following units:

Name of Ship.	Built at.	Engines Built by.	Speed.
Lion	Devonport	Vickers	28.5
Tiger	Clydebank	J. Brown	28
Princess Royal	Barrow	Vickers	28.5
New Zealand	Govan	Fairfield	25
Indomitable	Govan	Fairfield	27.3

As the speed of a squadron is that of its slowest ship, the German force was seriously handicapped owing to the presence of the *Blucher.* The immunity which had been enjoyed on the two previous occasions had apparently led the Germans to anticipate that on January 24th the raiding force would again not encounter any vessels of the British Fleet, with heavy guns, which could steam faster than the *Blucher.*

The German ships, when sighted, were approximately 14 miles east-south-east of the British force. Sir David Beatty steered south-east with a view to securing the lee position, and, if possible, cutting the enemy off from his base. The British admiral states that "speed was worked up to 28 and 29 knots, and the enemy were gradually being overhauled." At about 18,000 yards "slow and deliberate fire" was opened, and it is added, "we began to hit at a range of 17,000 yards." The *Tiger,* as was, perhaps, to have been expected, and the *Lion,* drew ahead of the

remainder of the squadron and, while themselves exposed to heavy fire, were able to punish the enemy severely. At last the German admiral decided to sacrifice the *Blucher*, and he returned to port at the highest speed, with two of his battle cruisers injured and heavily on fire, although apparently their stability was not seriously affected by any waterline hits. The *Lion*, on the other hand, was penetrated at the waterline by an 11-inch shell and, owing to her condition and the presence of the enemy's submarines, the action was broken off.

In view of the character of the two squadrons engaged, the result of the action is illuminating. If the German admiral, as was apparently his decision, determined not to stand by the *Blucher* and not to fight, he ought to have been able, on the trial speeds of his three battle cruisers, to return to port without being hit by a single British shell. On paper his three ships were at least as fast as, if not in one case faster, than the British vessels. Before the war occurred any German officer would have supported the contention that the three German ships could show their heels to the British vessels. In practice this proved not to be the case. Sir David Beatty's report goes to show that the *Lion* and *Tiger* gained rapidly on the three opposing battle cruisers and were thus able to bring them within effective gun fire and do them considerable damage. Owing to the principles in design for which Lord Fisher was responsible the British ships mounted thirty-two 13.5-inch guns and sixteen 12-inch guns. On the other hand the German squadron mounted eight 8.2-inch guns, twenty-nine 11-inch and eight 12-inch. In retreat the Germans were able to bring into action about the same number of weapons as the British ships could train upon them, but the latter possessed the more deadly weapons.

The action was a triumph of the all big-gun principle, but it was no less a triumph for the British engineering firms concerned in the construction of the vessels, and for the engineer officers and their staffs. If the British ships, in the emergency, had not realized their speed expectations, the action would have been inconclusive. The chase was a stern one between two opposing squadrons, with no great disparity in their paper speeds; the British engines and boilers in the hour of trial proved superior to the German engines and boilers. The engagement forms an interesting commentary upon the boastful statements in which German firms indulged during peace, and exhibits in the eyes of the world the absurdity of the allegation put forward by Krupps to the effect that the British wire-wound gun was a weak and unreliable weapon.

What other type of ship but the battle cruiser could have brought Admiral Graf von Spee's squadron to action off the Falkland Islands? The enemy force by superior speed and superior gunfire had sunk the weaker British cruisers, *Good Hope* and *Monmouth*, off Coronel, on November 1st. On December the 8th four of the five enemy ships, under the German Admiral, were sunk off the Falkland Islands. Between these two dates the British Admiralty had carried out a concentration movement unknown to the enemy. In the time available battleships could have steamed from a British home port to the Falkland Islands, and could have reached Port Stanley on December 7th. The Admiralty, however, were aware, on the one hand, that they possessed no battleships with a speed of more than about 21 knots, and that the German Admiral had under his flag the *Scharnhorst*, with a speed of 22.5, and the *Gniesenau*, which on her trials attained a speed of 23.8 knots. Associated with this squadron were three light cruisers, the *Leipzig* of 23 knots, the *Nurnberg* of 23.5 knots, and the *Dresden* of 27 knots. It must have been apparent, therefore, that battleships of 21 knots could not overhaul the German ships, and that something more speedy was necessary if that object was to be attained. Under the British flag there were light cruisers which could steam faster than the German flagship and her armored consort,

but no such vessel carried anything heavier than the 6-inch gun, while the two principal German ships were armed with sixteen 8.2-inch weapons.

In these circumstances the British Admiralty decided that if the German force was, first, to be overhauled and then to be defeated, the ideal type of vessel for the purpose was the battle cruiser. The naval authorities, therefore, despatched from England, under Vice-Admiral Sir Doveton Sturdee, the *Invincible* and the *Inflexible*, with a nominal speed of 26 knots and each mounting eight 12-inch and sixteen 4-inch guns, in association with armored belts of a maximum thickness of 7 inches—a thickness slightly in excess of that possessed by the two principal German vessels. The *Invincible* was completed at Elswick in 1909, being provided with Parson's turbines by Messrs. Humphrys and Tennant; the *Inflexible* was constructed at Clydebank, and was engined by Messrs. J. Brown and Co. Providing no serious breakdown occurred in the ships under the White Ensign, the result of an encounter with Graf von Spee's squadron was inevitable. In the event, British engines maintained their high reputation and British engineers and their staffs showed how efficiently they could be worked. The two large ships of the German squadron were sunk, and, owing to the foresight exhibited by the British Admiralty, two of the three light German cruisers were also destroyed—the *Dresden*, the swiftest of the trio, alone escaping, which shows that the enemy found superior speed in this case no mean asset. The success of this rounding-up movement was due to the orders which resulted in the British cruisers *Carnarvon*, *Cornwall*, *Kent*, *Glasgow* and *Bristol*, being concentrated on the Falkland Islands, together with the auxiliary ship, the *Macedonia*, for the decisive event. Neither of the three British armored cruisers was of recent construction, as the following statement shows:

Name of Ship.	Built at.	Engines built by.	Speed.	Completed.
Carnarvon	Beardmore	Humphrys.	23.3	1905.
Cornwall	Pembroke	Hawthorn.	23.68	1904.
Kent	Portsmouth	Hawthorn.	21.7	1903.

These three ships, together with the *Glasgow* and *Bristol* (built at Govan and Clydebank, and engined by Fairfield and Messrs. John Brown and Co., respectively), amply justified the confidence of their builders, in spite of the years which had elapsed since they were completed.

The story of the *Kent's* pursuit of the *Nurnberg* is a romance of marine engineering. The story has been told by Midshipman John Esmonde in a letter to his father, Sir Thomas Esmonde, M.P.

The *Kent*, a 21-knot cruiser, was ordered to chase the *Nurnberg*, a 25-knot ship and also a much more modern one than the *Kent*. She had only a few hundred tons of coal on board to catch the *Nurnberg* with. The old *Kent* set off, and they worked up to 22, more than she had ever done on trials. Then the word was passed up that there was hardly any coal left. "Well," said the Captain, "have a go at the boats." So they broke up all the boats and smeared them with oil and put them in the furnace. Then in went all the arm chairs from the wardrooms, and then the chests from the officers' cabins. They next burnt the ladders and all—every bit of wood was sent to the stokehold. The result was that the *Kent's* speed became 24 knots, and she caught the *Nurnberg*, and after a stiff fight, in which several men were killed, the *Nurnberg* was sunk.

The performance of the *Kent* on this occasion will not soon be forgotten and, when it is recalled, tribute will be paid not only to the firm responsible for the engines, but to Engineer-Commander George E. Andrew, Engineer Lieut.-Commander Alfred E. E. Rayner, Engineer Lieut. (retired) Victor O. Foreman, and the engine-room staff, who must have put forth almost superhuman efforts to achieve so notable a success.

The battle cruiser has come to stay. It represents an effective compromise in warship design, particularly for a rich nation, with world-wide, home, commercial and vast territorial interests, determined to exercise command of the sea in time of war. High speed enables a concentration to be carried out swiftly and confers no mean tactical advantages when face to face with the enemy, while the association of high speed with an armament not inferior in caliber to that of a battleship contributes, as experience has shown, to decisive results, at a slight cost in casualties. The battle cruiser is "not dead"; there is certainly no intention "to inter it away from the public gaze." The tendency of development of this type of ship is revealed in the vessels of the *Kongo* and *Queen Elizabeth* classes, which are now passing, or have passed, under the Japanese and British flags. The *Kongo*, like the British ships of the *Queen Elizabeth* type, mount eight big guns only, but, whereas the Japanese ship carries the 14-inch weapon of the Vickers type, the British ships have the new 15-inch gun. Everything suggests that the future armored ship will consist of a compromise in which speed and gunpower will be further emphasized.—"Cassier's Engineering Monthly."

SOME TESTS MADE AT THE ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MARYLAND.

FROM REPORTS PREPARED BY D. J. MCADAM, CHEMIST.

BOILER TUBES.

1. Thirteen boiler tubes were selected at random from about 200 tubes that had to be cut out of boiler drums intended for the *Jacob Jones* and *Wainwright*. The ends of the tubes had split when they were expanded and belled; about 200 out of 9,300 of the tubes proved defective.

2. The tubes were manufactured by the Pittsburgh Steel Products Company. Since the manufacturers could not trace the material back to the heat numbers of the steel, the results of their chemical analysis could not be obtained.

3. The purpose of the test was to make a chemical and metallographic examination of the samples received in order to determine the quality of the metal.

4. The tubes, after being photographed, were each cut longitudinally into two pieces for metallographic examination. Samples for chemical analysis were also taken. The arrangement of the metallographic specimens, numbers from 1 to 26, is as shown in Plate I. After the metallographic examination, ductility tests were made on pieces in their original condition and on pieces heat treated at this station.

5. The results of the chemical analysis were as follows:

	No. of Determinations.	*Average.*
Carbon	2	0.166
Sulphur	1	0.029
Phosphorus	2	0.042
Manganese	1	0.392
Silicon	1	0.037

The results of the metallographic examination are illustrated by the photomicrographs on cards 1 to 18. In all these photomicrographs, the white areas represent ferrite and the dark areas represent pearlite.

Card 1 shows the outer surface of specimen 1 near the crack shown on

Plate I. The magnification is 100. In this region the ferrite grains are elongated; the pearlite also occurs in streaks extending lengthwise with the tube. The quantity of pearlite varies greatly in different regions on the surface of the tube.

Card 2 shows the outer surface of specimen 2 near the crack shown on Plate I. The magnification is 500. The pearlite, which is less in quantity than in the area shown on card 1, occurs in segregated regions surrounded by relatively large regions of pure ferrite. In such a structure, the brittle high-carbon regions would behave to a certain extent like slag or other foreign particles. The pearlite in this condition has very little strengthening effect on the metal.

Card 3 shows the outer surface of specimen 4 at a magnification of 100. In this specimen the pearlite is unevenly distributed and occurs in irregular masses extending lengthwise with the tube.

Card 4 shows a longitudinal section of specimen 5 at a magnification of 100. The ferrite grains are distorted and the pearlite is very unevenly distributed.

Cards 5 and 6 show the outer surface of specimen 8 at magnifications of 45 and 100, respectively. The ferrite grains are distorted and the pearlite occurs in streaks extending lengthwise with the tube. A broad streak of nearly pure ferrite containing slag particles is seen in the center of the photomicrographs.

Card 7 shows a longitudinal section of specimen 10 at a magnification of 100. On this surface the distribution of ferrite and pearlite is fairly uniform and there is no distortion of the ferrite.

Card 8 shows a transverse section of specimen 11 at a magnification of 45. Alternate layers of ferrite and pearlite extend around this section of tubing. This segregated structure is due to cold working followed by insufficient annealing.

Cards 9 and 10 show two views of specimen 13. Card 9 shows a longitudinal section at a magnification of 45. Pearlite streaks and banks of nearly pure ferrite are seen in this photomicrograph. Card 10 shows the outer surface of the same specimen at a magnification of 100. The area photographed is in line with a crack shown on Plate I. Much segregation of ferrite and pearlite is found in this specimen; regions of more than average carbon content are surrounded by areas of nearly pure ferrite. In the streaks of ferrite are a few slag particles.

Card 11 shows the outer surface of specimen 16 at a magnification of 100. Though the structure of this specimen is better than that of some of the other specimens examined, there is considerable segregation of pearlite. The pearlite is found in small masses unevenly distributed and surrounded by large areas of nearly pure ferrite.

Card 12 shows the outer surface of specimen 17 at a magnification of 100. The ferrite grains are elongated in the direction in which the tube was drawn; the pearlite also occurs in elongated masses.

Card 13 shows the outer surface of specimen 20 at a magnification of 100. In this specimen there is some distortion of the ferrite grains, they are elongated in the direction in which the tube was drawn. The pearlite is small in amount and unevenly distributed.

Card 14 shows the outer surface of specimen 22 at a magnification of 100. The area here shown is near a crack in the metal. In this specimen there is much segregation of pearlite. Surrounding a region of nearly pure ferrite, seen in the central part of the photograph, are areas having excess of pearlite. A decided streak of pearlite is visible in the upper part of the photograph.

Card 15 shows a longitudinal section of specimen 23 at a magnification of 100. Elongation of the ferrite grains in the direction in which the tube was drawn is noticeable in this section. The pearlite also occurs in longitudinal streaks.

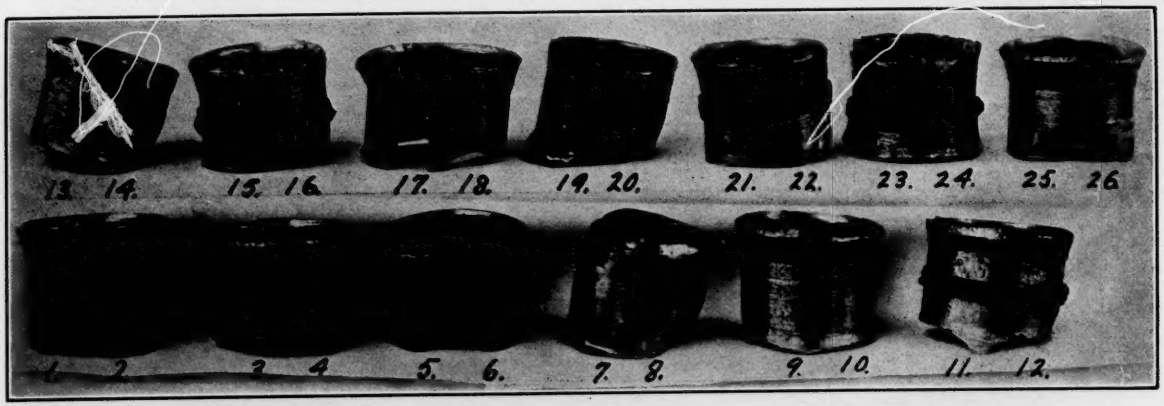


PLATE I.

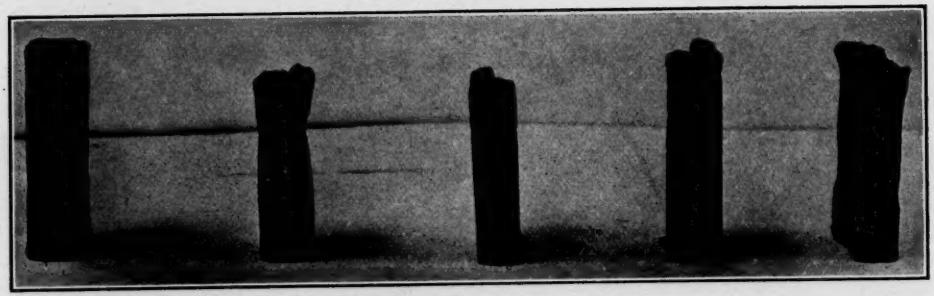
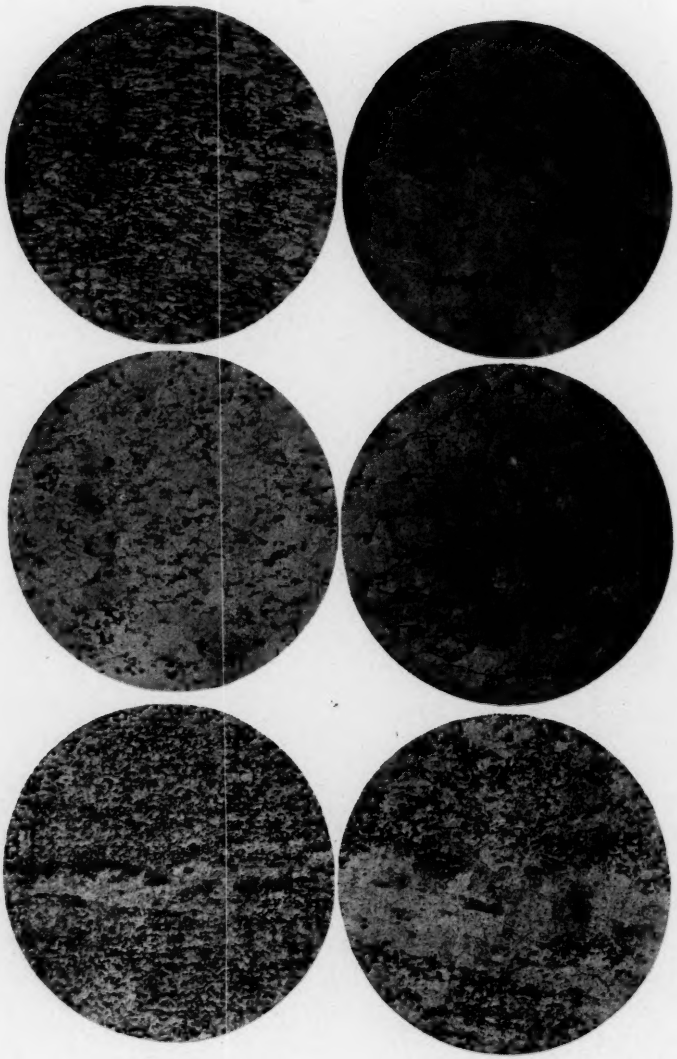
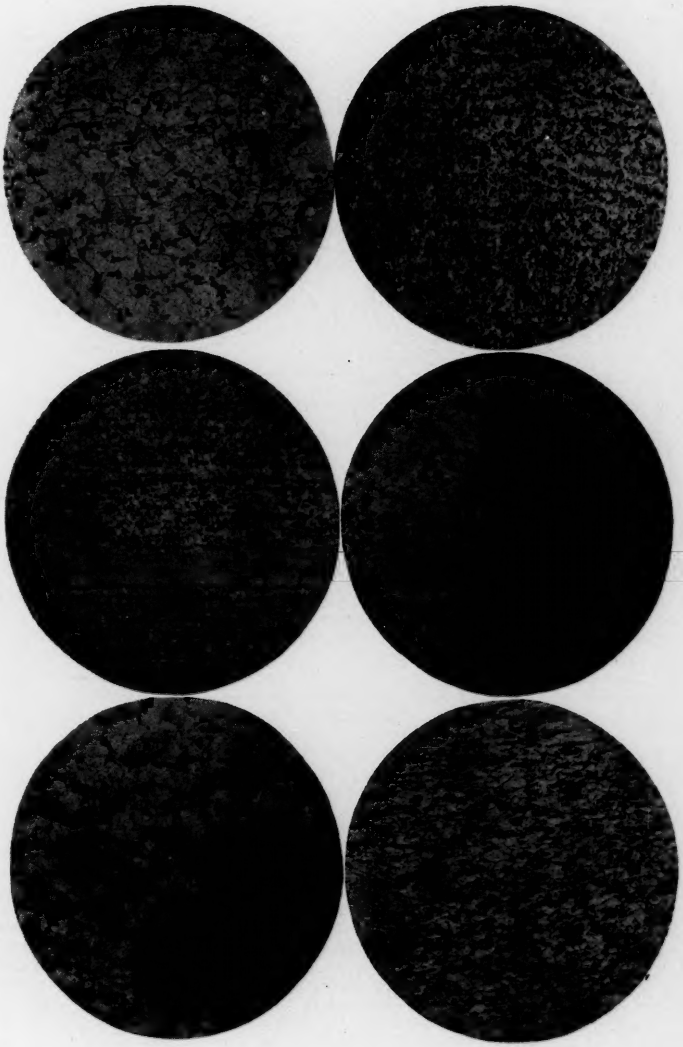


PLATE II.



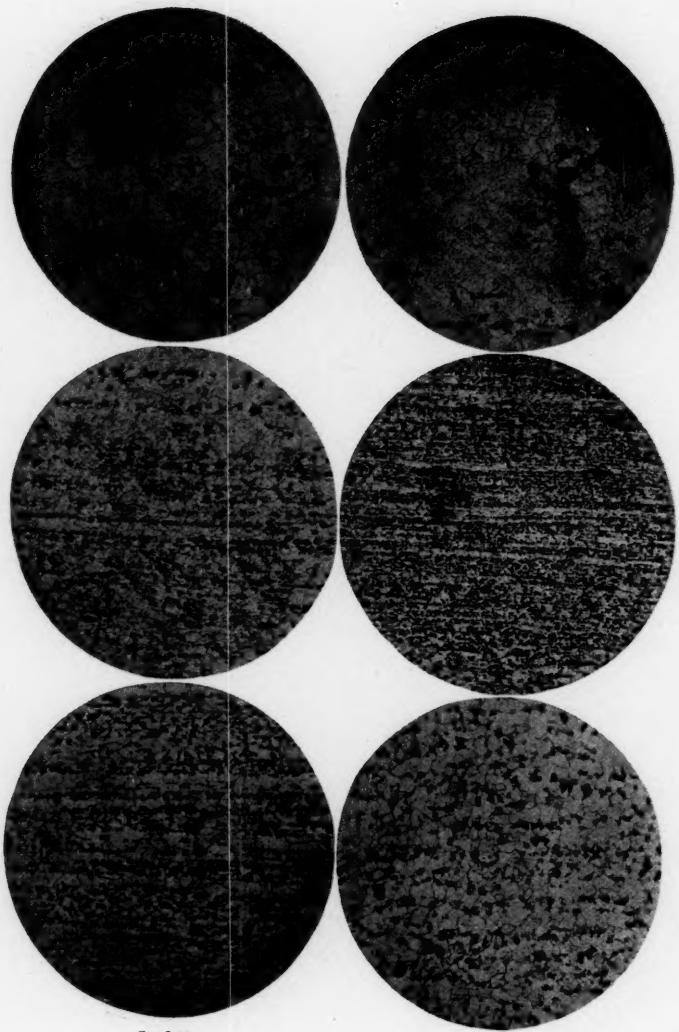
Card 1.
3.
5.

Card 2.
4.
6.



Card 7.
9.
11.

Card 8.
10.
12.



Card 13.
15.
17.

Card 14.
16.
18.

Cards 16 and 17 show a longitudinal section of specimen 25 at magnifications of 45 and 100, respectively. Streaked segregation of ferrite and pearlite is prominent in this specimen. Distortion of the ferrite grains is also noticeable.

Card 18 shows a transverse section of specimen 11 after heat treatment at the Station. It was heated at 1,600 degrees F. for two hours and cooled in the furnace. The original structure was as shown on card 8. Though some segregation still remains, the heat treatment has removed most of the effects of cold working. Pronounced streaks are very difficult to remove by heat treatment.

6. The results of one of the ductility tests, which the Bureau of Steam Engineering specifies for the inspection of boiler tubes, are shown on Plate II. Two pieces of the metal in its original condition were flattened in a vice; the results are shown on the left of the plate. Three similar pieces were then heated in a furnace at 1,600 degrees F. for two hours, cooled in the furnace and flattened in the vice; the results are shown on the right of the plate. It will be noticed that the two pieces on the left both show cracks at the point of greatest bend, while the three heat-treated pieces show no cracks.

DISCUSSION.

The chemical analysis shows that the composition of this material may be considered satisfactory, although it would be better if the percentages of carbon and phosphorus were somewhat lower. The Bureau of Steam Engineering specifies no percentage limits for these constituents in boiler tubes. The American Society for Testing Materials, however, in specifications adopted June 1, 1912, gives 0.15 and 0.04, respectively, as the highest allowable percentages of carbon and phosphorus. These limits are reached or slightly exceeded in the tubes under discussion. Since both carbon and phosphorus decrease the ductility of steel, evidently the phosphorus content should be kept as low as possible and the carbon content should be kept as low as is consistent with the strength required.

As a result of the metallographic examination, it is evident that most of the specimens examined show distortion of the ferrite grains and more or less decided segregation of ferrite and pearlite. In some regions the pearlite occurs in scattered masses, elongated in the direction in which the tube was drawn; in other regions there are well-defined alternating streaks of ferrite and pearlite. The quantity of pearlite also varies considerably in different parts of the tubes.

The grain distortion and much of the streaked segregation are evidently due to cold working followed by insufficient annealing. Some of the segregation, however, must undoubtedly be traced back to segregation in the billets from which the tubes were made. Careful annealing will remove grain distortion and much of the streaked segregation; broad streaks cannot be removed by any practicable heat treatment.

The effects of cold working on the physical properties of steel are well known. The brittleness of these tubes, as shown by their behavior when the ends were expanded, is evidently due to the distorted and segregated structure caused by cold working. These conclusions, reached as a result of the metallographic examination, are confirmed by the results of the physical tests described in substance in the foregoing. According to the specifications of the Bureau of Steam Engineering, "Attest piece cut from each end of each tube must stand flattening under a press or hammer * * * without showing any defects on the outside." By reference to Plate II, it will be noticed that the two pieces on the left of the plate did not meet these requirements; the three heat-treated pieces, however, were flattened without showing any defects on the outside. It seems evident, therefore, that if these tubes had been more thoroughly annealed their brittleness would have been removed and the ends could have been expanded without cracking.

CONCLUSIONS.

The chemical analysis has shown that the composition of this material may be considered satisfactory, although it would be better if the percentages of carbon and phosphorus were somewhat lower.

Nearly all of the metallographic specimens have shown grain distortion and more or less decided segregation of ferrite and pearlite. This structure is caused by cold working followed by ineffective annealing. Since such treatment causes brittleness, the brittleness of these boiler tubes is therefore explained. The fact that samples from the tubes, when carefully annealed at the Experiment Station, lost most of their brittleness, lends additional support to this explanation.

HULL PLATE FROM DESTROYER.

A metallographic examination of three samples from a hull plate on one of the recent destroyers was made. The plate had failed on the trial trip of the destroyer. It cracked in a transverse direction and partly opened up. The percentages of sulphur and phosphorus in the metal are not given and no special heat treatment was given this metal.

The purpose of the test was to make a metallographic examination of the metal and determine its quality.

Plate I shows a photograph of the three steel samples. Samples marked 5, 6 and 12 were then cut into smaller pieces and polished on various surfaces for metallographic examination. Plate II is a photostat copy of a sketch showing the location of these metallographic specimens. Thirteen photomicrographs showing the typical structure of the material are shown on cards 1 to 13.

RESULTS.

In the photomicrographs of etched specimens, the white areas represent ferrite and the dark areas represent pearlite.

Card 1 shows the unetched surface of specimen 4, at a magnification of 45. Many slag particles are scattered over the surface. Some of the slag particles occur in irregular lines extending in the direction in which the plate was rolled.

Card 2 shows an edge of specimen 5 at a magnification of 100. A fine granular structure is found in this specimen. There are thin streaks of ferrite and pearlite extending in the direction in which the plate was rolled; these streaks, however, being very slight, would not injure the properties of the metal.

Card 3 shows the structure of specimen 7 at a magnification of 100. A fine-grained, uniform structure is found in this specimen.

Card 4 shows the unetched surface of specimen 9 at a magnification of 45. The area shown is near the surface of the break. A crack, extending from the surface of the break, appears in this photograph; a short crack nearly parallel to the longer one is also seen. These fine cracks follow lines of segregation of slag particles.

Cards 5 and 6 show adjacent areas on the unetched surface of specimen 12. The magnification in each case is 45. On card 6 the line of break is visible in the upper part of the photograph. Many small slag particles are scattered over the surface of the metal. Groups of blow holes, with cracks passing through them, also occur in this specimen; the cracks are parallel to the general direction of the line of break.

Card 7 shows a portion of the same view that is shown on card 6. The magnification is 100. Large blow holes, some of which are starting points for cracks, are plainly visible in this photograph.

Cards 8 and 9 show the etched surface of specimen 12 adjacent to the surface of the fracture; the area includes part of that shown on card 5. Card 9 shows the central part of the area shown on card 5. The

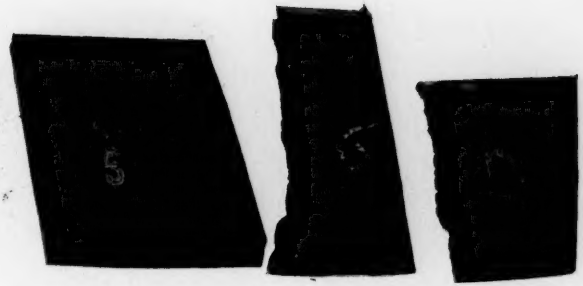
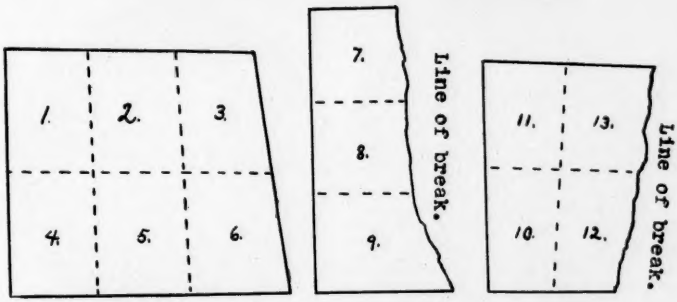


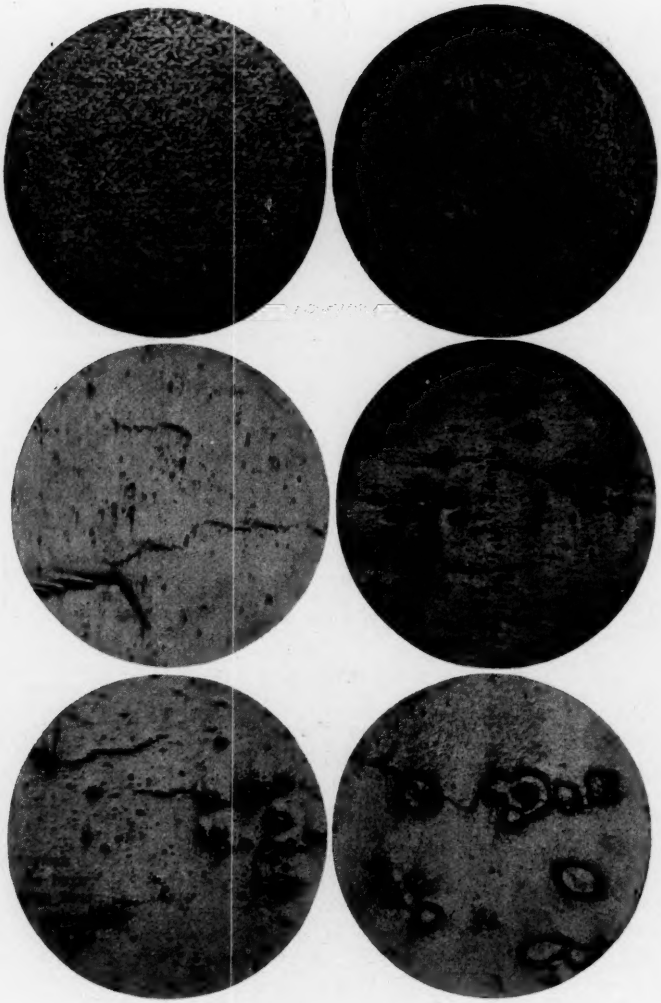
PLATE I.



Sample 5. Sample 6. Sample 12.

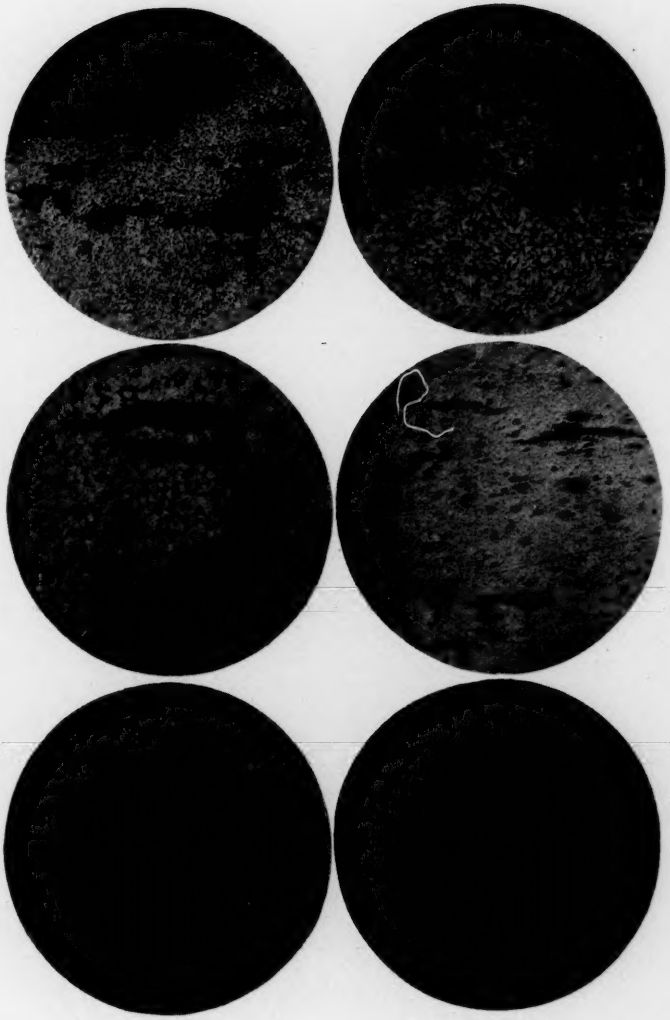
PLATE II.

Card 1.—Unetched.



Card 2.
4.—Unetched.
6.—Unetched.

Card 3.
5.—Unetched.
7.—Unetched.



Card 8.
10.
12.

Card 9.
11.—Unetched
13.

general structure of the metal is fine grained and uniform. Many blow holes, however, are visible on the surface; extending through these are incipient cracks parallel to the general direction of the surface of fracture.

Card 10 shows a portion of the etched surface of specimen 12 about ¼ inch from the surface of fracture. The magnification is 200. Two cavities in the metal are shown in this photomicrograph. Both of these cavities are probably blow holes, although one of them might be a hole left by the removal of slag during polishing. Minute cracks originate at both of the holes.

Card 11 shows the unetched surface of specimen 13 about ⅛ inch from the break. The magnification is 45. Many small slag particles and a few blow holes are scattered over the surface of the metal. Incipient cracks are visible starting from blow holes and slag particles and extending parallel to the surface of the fracture.

Cards 12 and 13 show the etched surface of specimen 13 about ½ inch from the surface of fracture. Card 13 shows, at a higher magnification, a portion of the same area that is seen on card 12. The etching has somewhat obscured the small slag particles, but groups of blow holes are visible. Some of the blow holes are starting points for cracks which are parallel to the surface of the fracture.

DISCUSSION.

The structure of the metal itself, as shown by the metallographic examination, is fine grained and fairly uniform. The edges of the plate show a slight degree of streaked segregation due to the rolling process; the amount of this segregation, however, is not sufficient to injure the quality of the metal.

Though the structure of the metal itself is good, non-metallic enclosures are present in such amounts that the material is decidedly weakened. As shown on cards 1, 4, 5, 6 and 11, many small slag particles are scattered throughout the metal. Many of these occur in lines or groups extended in the direction in which the metal was rolled. That these slag groups are regions of weakness is shown by the microscopic cracks which frequently follow the lines of segregation.

The chief source of weakness in this material, however, is the presence of groups of blow holes. These are especially numerous near the surface of break. Many incipient cracks, starting from these blow holes and often following an irregular line of blow holes, are found in this material. These cracks are invaribly parallel to each other and to the general direction of the surface of fracture.

CONCLUSIONS.

Though the general structure of the metal is good, the material is much weakened by the presence of non-metallic enclosures. Many fine particles of slag are scattered throughout the metal, and are often found in groups which are regions of weakness. The chief defect in the material, however, is the presence of groups of blow holes. These blow holes are especially numerous near the surface of fracture and were undoubtedly the chief cause of the failure of the plate.

—

U. S. S. *TACOMA*, PROPELLER SHAFT.

Examination was made of a section from the broken propeller shaft. The shaft had broken just abaft the after strut, at the end of the composition sleeve and the beginning of the taper. The section received for examination included the face of the fracture; this section including the composition sleeve is shown in Plates I and II. The appearance of the section after removal of the sleeve is shown in Plate III.

This shaft was manufactured by the Bethlehem Steel Company under specifications requiring high grade material. It was forged by press from a 35-inch round, fluid-compressed ingot; the shaft was annealed, tempered and then twice annealed. Details of heat treatment are not available.

The purpose of the test was to make a thorough examination of the material in order to determine its quality and, if possible, the cause of the fracture.

After the photographs (Plates I and II) were taken the composition sleeve was removed and the material was cut into sections as indicated by the dotted lines on Plate III. Samples for chemical analysis were taken and pieces were cut for physical tests and metallographic examination. A number of specimens were heat treated at the Experiment Station and subjected to physical tests. Sulphur prints were also made and a photograph of one of these is included in this report as Plate VI.

Results.—The results of the chemical analysis were as follows:

	Number of determinations.	*Average percentage.*
Carbon	2	0.348
Sulphur	2	0.0314
Manganese	2	0.724
Silicon	1	0.265
Phosphorus	1	0.021
Nickel	2	3.22

The results of tensile, bending, torsion, impact and endurance tests are given in Tables I to V. Both longitudinal and transverse test specimens were used. Some of these were heat treated at the Experiment Station. The heat treatment used was as follows:

Heat treatment (1).

Heated slowly to 1,700° F.; maintained for 1 hour; quenched in oil.
Reheated slowly to 1,425° F.; maintained for 1 hour; quenched in oil.
Reheated slowly to 1,125° F.; maintained for 1 hour; cooled in furnace.

Heat treatment (2).

Heated slowly to 1,700° F.; maintained for 1 hour; quenched in oil.
Reheated slowly to 1,425° F.; cooled in furnace.

The physical test specimens are designated by a letter showing the section from which the specimens were taken as seen on the photograph, Plate III.

The specimens used for metallographic examination were as follows:

Section A, 24 specimens.
Section B, 34 specimens.
Section C, 9 specimens.
Section D, 50 specimens.

Of these specimens, 13 photomicrographs were made showing the typical structure. In the six photomicrographs showing etched surfaces the white areas represent ferrite and the dark areas sorbite.

Cards 1 and 2 show two views of a longitudinal surface of specimen A17. The magnification in each case is 45. On the unetched surface are many slag particles arranged in lines extending lengthwise with the shaft. On card 1 a region near the fracture is shown; on the surface is a crack which extends away from the fracture.

Cards 3 and 4 show two views of a longitudinal surface of specimen A21 at a magnification of 45. Groups of many slag and sulphide particles are visible on the unetched surface. On card 3, which shows a region at the line of break, it will be noticed that the slag groups are especially numerous along the line of fracture.

Card 5 shows a longitudinal surface of specimen B30 at a magnification of 45. On the etched surface is a moderate sized network consisting of

PLATE I.

PLATE II.

PLATE III.

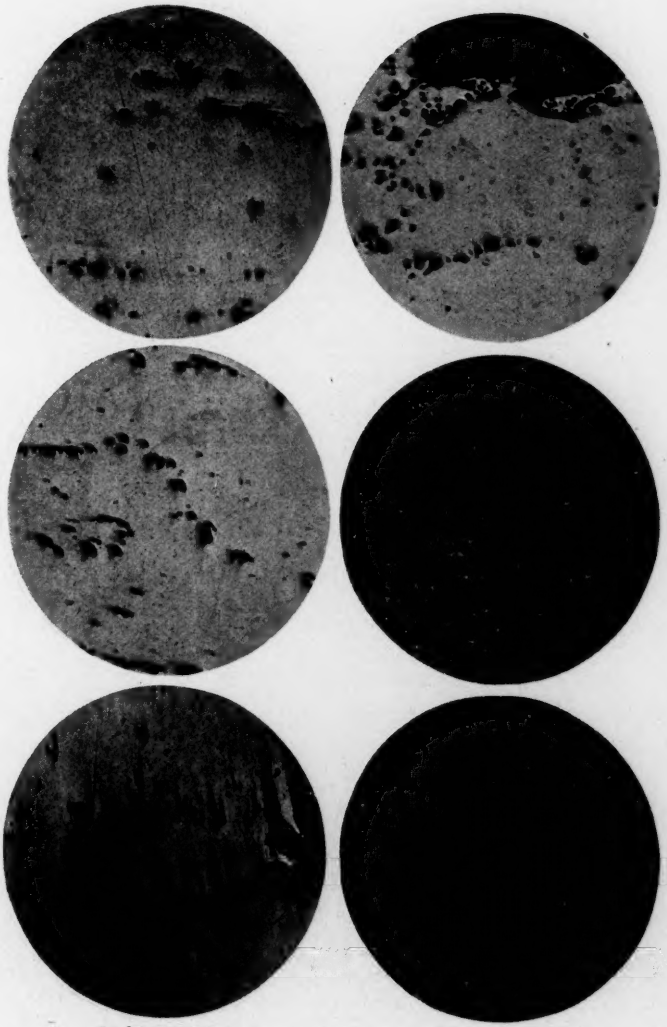
PLATE IV.

PLATE V.

U. S. S. Tacoma.—Propeller Shaft.

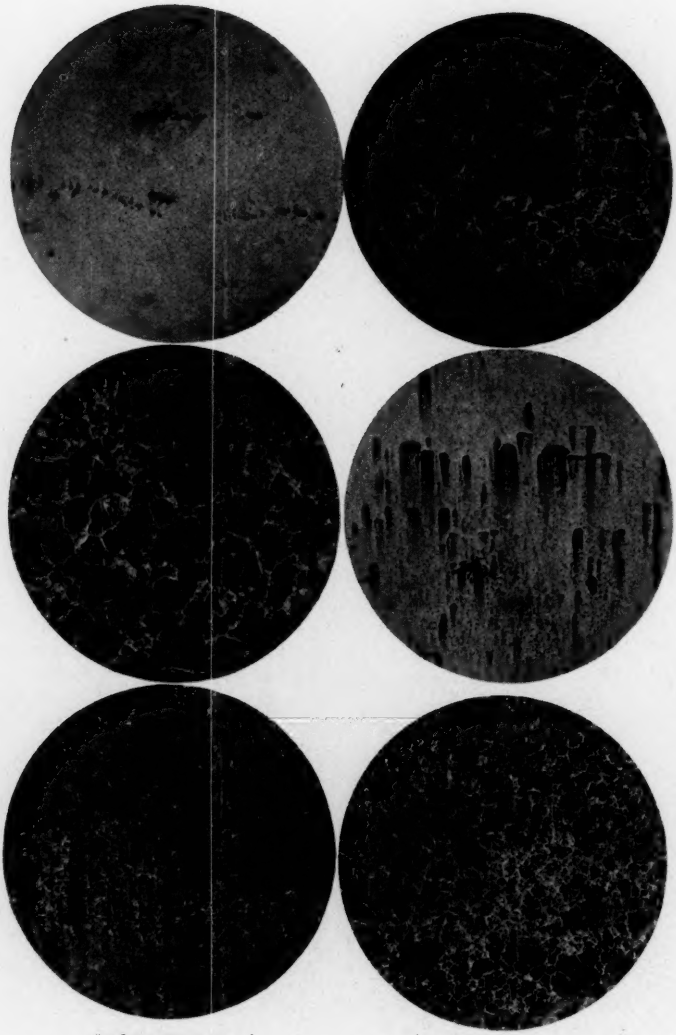
PLATE VI.

Card 1.—Unetched.



Card 2.—Unetched.
4.—Unetched.
6.—Unetched.

Card 3.—Unetched.
5.
7.



Card 8.—Unetched.
10.
12.

Card 9.
11.—Unetched.
13.

ferrite and sorbite. The view here shown is about two inches from the surface of break. In the middle of the photograph is a longitudinal streak lighter than the rest of the surface and containing many elongated particles of slag and manganese sulphide. Evidently the carbon content is below the average in this streak. Many such streaks were noticed in this material.

Cards 6 and 7 show two views of a longitudinal surface of specimen B34. The magnification in each case is 45. Card 6 shows the unetched surface at the line of break. Lines of slag particles, such as are seen on cards 1 to 4, are visible on the surface of this specimen. A short crack extends from the surface of break into one of these slag lines. On card 7, which shows the surface of the same specimen after etching, a broad streak having less than the average carbon content is shown. The structure is like that seen on card 5. In the streak are groups of slag particles.

Card 8 shows a longitudinal surface of specimen D35 at a magnification of 45. The view here shown is distant from the fracture. Lines of many small slag particles are seen on this surface.

Cards 9 and 10 show etched surfaces of specimens D38 and D46. The magnification in each case is 100. As on cards 5 and 7, a moderate-sized network is here shown. On card 9 a few elongated slag particles are visible.

Cards 11 and 12 show two views of a longitudinal surface of specimen D49. In both of these photomicrographs many segregated slag particles are visible. They show more clearly, however, on the unetched surface as seen on card 11. On card 12 it can be seen that there is segregation of ferrite and sorbite corresponding to the slag segregation.

Card 13 shows a longitudinal surface of specimen D50 at a magnification of 45. A longitudinal streak of less than average carbon content is seen in this specimen. This streak, with its lines of slag, is similar to those shown on cards 5 and 7 and to many other streaks found in this metal.

DISCUSSION.

As shown on cards 5, 7, 9, 12 and 13 the average grain size of this material is moderate. The metal, however, as shown on cards 1, 2, 3, 4, 6, 8 and 11, contains much slag. The slag particles are arranged in lines extending lengthwise with the shaft; they are especially numerous near the surface of the break, and, as shown on card 3, the fracture often follows these lines of segregation. That these slag streaks contain considerable manganese sulphide is indicated by Plate VI, which is a photograph of a sulphur print. Many lines of sulphid are visible on this print; their direction is lengthwise with the shaft.

Corresponding to the slag streaks are many broad bands having less than the average carbon content; a good example of such banks is shown on card 13.

Such segregation of slag and carbon is undoubtedly a source of weakness in the metal. It should not be assumed, however, that this alone caused the fracture of this shaft to take place. The quality of the metal is only one of a number of factors on which the endurance of the shaft depended. Plates 3, 4 and 5 show that the break originated at a hole in the shaft which was tapped out to receive a set screw. Ridges on the fractured surface radiate from the bottom of this screw hole, showing clearly that the break started from this point. Several examples of fractures, passing through such screw holes in shafts have been received at the Experiment Station. The weakening of the shaft by these holes is out of all proportion to the amount of metal removed; they should, therefore, be avoided whenever possible.

To determine the physical qualities of this metal a number of physical tests were made, the results being given in the foregoing. Tensile, bend-

ing, torsion and impact tests, made as usual with longitudinal specimens, are given in Tables I, III, IV and V. It will be seen that these longitudinal specimens gave very good results when subjected to all the tests except the single-impact tests given in Table V. The values obtained by the single-impact tests are less than they should be for a material of this composition; the metal, even after annealing, shows brittleness when subjected to sudden shock.

It is not customary to make physical tests with transverse specimens from shafts. Yet in shafts the transverse strength and ductility of the metal are of importance, though of less importance than in ordnance. The relation between the physical properties of longitudinal and transverse specimens varies considerably with the composition of the metal and the treatment it has received. It was thought, therefore, that valuable results might be obtained by making physical tests on transverse specimens from this shaft.

As shown in Table I, the transverse specimens had much less elongation and reduction of area than the longitudinal specimens, while the tensile strength and elastic limit are about the same. The impact test given in Table V also showed some superiority of the longitudinal over the transverse specimens. Part of the inferiority of the transverse specimens is undoubtedly due to the longitudinal streaks of slag and sulphide. Even with very uniform material, however, there is often a great inferiority shown by transverse specimens.

Two kinds of heat treatment were used, the results of which have been given in the foregoing. In both of these a preliminary heating at 1,700 degrees was given to produce uniformity. In heat treatment (1) the process known as "double annealing" was used, while in heat treatment (2) the material was simply annealed and cooled slowly. It will be noticed in Table II that neither kind of heat treatment produced improvement in the transverse tensile specimens. In the longitudinal specimens, however, the yield point was increased about 50 per cent., the tensile strength and ductility remaining about the same. As shown in Table V, heat treatment (1) tripled the numerical values obtained with longitudinal impact specimens and more than doubled the values obtained with transverse specimens; heat treatment (2), however, had very little effect on either longitudinal or transverse specimens. These results seem to show the superiority of "double annealing" over single annealing or "softening." It is not claimed, however, that this superiority would be found in an equal degree after the heat treatment of large objects.

More data as to the relation between the physical properties of longitudinal and transverse specimens, for different kinds of steel, will be obtained as opportunity arises. The relation between the physical properties produced by different kinds of heat treatment will also be the subject of investigation when possible.

CONCLUSIONS.

As shown by the metallographic examination, the grain size of this metal is moderate and the distribution of ferrite and sorbite is fairly uniform. There is, however, much slag and manganese sulphide segregated in groups and in lines extending lengthwise with the shaft. These slag particles are especially numerous near the surface of the break, and the fracture often follows the lines of segregation. Corresponding to these slag lines are many broad streaks having less than the average carbon content. This segregation of slag and carbon is undoubtedly a source of weakness in the metal.

In physical tests, longitudinal specimens, with the exception of impact specimens, gave good results. The impact specimens gave results which are too low for material of this composition; the metal, therefore, displays some brittleness toward sudden shock. Transverse tensile specimens

were deficient in ductility. Transverse impact specimens also showed much brittleness.

When heat treated at the Experiment Station longitudinal tensile and torsion specimens showed much increase in elastic limit alone, while transverse tensile specimens showed little or no improvement of any kind by heat treatment. Impact specimens showed much improvement by "double annealing" but practically no improvement by single annealing or "softening." It seems evident, therefore, that the brittleness toward sudden shock is not due to "hardness" of the metal.

Though the metal has much slag segregation, and displays brittleness, particularly in a transverse direction, it should not be assumed that the break was entirely due to the defects of the metal. The screw hole shown in Plates 2, 3 and 4 undoubtedly weakened the shaft somewhat. It is impossible, therefore, to say whether or not the shaft would have broken if the metal had been better and other conditions had remained the same. It is equally impossible to say whether or not the shaft would have broken if the screw hole had not been there and other conditions had remained the same.

TABLE I.

TENSILE TESTS.—PROPELLER-SHAFT FORGING, U. S. S. "TACOMA."

	Longitudinal.			*Transverse.*			
Standard Specimen No.	E2	E7	Ave.	B2	D2	D4	Ave.
Diameter:							
Test piece, in.	.505	.505		.505	.505	.505	
Fracture, in.	.352	.355		.458	.463	.432	
Area:							
Test piece, sq. in.	.200	.200		.200	.200	.200	
Fracture, sq. in.	.097	.099		.164	.168	.146	
Reduction, pr. c.	51.5	50.5	51.0	18.0	16.0	27.0	20.3
Length between punch marks in inches	2.	2.		2.	2.	2.	
Elongation:							
Inch	.47	.47		.17	.31	.36	
Per cent.	23.5	23.5	23.5	8.5	15.5	18.0	14.0
Yield Point:							
Pounds	11,300	13,100		12,150	12,700	12,800	
Pounds per sq. in.	56,500	65,500	61,000	60,750	63,500	64,000	62,750
Maximum Stress:							
Pounds	21,080	21,060		20,100	21,200	21,280	
Pounds per sq. in.	105,400	105,300	105,350	100,500	106,000	106,400	104,300
Remarks:							
Fracture	medium	medium		coarse striated	coarse striated	coarse striated	

TABLE II.

TENSILE TESTS.—PROPELLER-SHAFT FORGING, U. S. S. "TACOMA."

Heat-Treated Specimens.

	Longitudinal.			*Transverse.*		
Standard Specimen No.	E11	B1	B3	D10	D3	Ave.
Diameter:						
Test piece, in.	.505	.505	.505	.505	.505	
Fracture, inch	.325	.460	.453	.430	.450	
Area:						
Test piece, sq. in.	.200	.200	.200	.200	.200	
Fracture, sq. in.	.083	.166	.161	.145	.159	
Reduction, per cent.	58.5	17.0	19.5	27.5	20.5	21.1
Length between punch marks in inches	2.	2.	2.	2.	2.	
Elongation:						
Inch	.46	.28	.20	.35	.24	
Per cent	23.0	14.0	10.0	17.5	12.0	13.4
Yield Point:						
Pounds	18,800	13,000	12,150	13,100	12,800	
Pounds per sq. in.	94,000	65,000	60,750	65,500	64,000	61,310
Maximum Stress:						
Pounds	23,250	21,000	18,950	21,100	21,100	
Pounds per sq. in.	116,250	105,000	94,750	105,500	105,500	102,690
Heat treatment	1	1	2	1	1	
Remarks:						
Fracture	fine	coarse striated.		coarse striated.		

TABLE III.

TORSION TESTS.—PROPELLER-SHAFT FORGING, U. S. S. "TACOMA."

	Longitudinal Specimens.	
Standard Specimen No.	E1	E12*
Diameter of piece, inch	.750	.750
Length between punch marks, inches	3.	3.
Angle of twist at elastic limit, degrees	1.63	2.2
Angle of twist at maximum torque, degrees	530	660
Elastic limit, pounds inches	3,200	4,400
Maximum torque, pounds inches	8,150	9,030
Shearing stress at elastic limit, pounds per sq. in.	38,600	53,100
Nominal shearing stress at maximum torque, pounds per sq. in.	98,400	109,000
Modulus of transverse elasticity, pounds per sq. in.	10,800,000	11,606,000

*Heat treated at the Experiment Station.

TABLE IV.

BENDING, ALTERNATING-IMPACT AND ENDURANCE TESTS.—PROPELLER-SHAFT FORGING, U. S. S. "TACOMA."

Bending Test:			
Standard specimen	E5		
Degrees of bending without breaking	180		
	Longitudinal,		
Alternating-impact test:			
Standard specimen	E4	E10*	E13*
Fall of hammer, centimeters	9	9	9
Number of falls required to break	507	632	643
Endurance test:			
Standard specimen	E3		E6
Load used, pounds	135		135
Fiber stress (353.38 times load) pounds	47,706		47,706
Ratio, fiber stress to elastic limit, per cent	78.2		78.2
No. of revolutions required to break	619,800		487,000

*Heat treated at the Experiment Station; Heat treatment (1).

TABLE V.

SINGLE-IMPACT TESTS.—PROPELLER-SHAFT FORGING, U. S. S. "TACOMA."

Direction.	Standard specimen ¼" × ⅜" cross section.	Energy required to break Ft. pounds.	Heat treatment.
Longitudinal	B4	24	Untreated
"	B5	27	"
"	B6	77	(1)
"	B7	76	(1)
"	C3	74	(1)
"	C4	78	(1)
"	C5	77	(1)
"	C6	81	(1)
"	C7	28	(2)
"	C8	31	(2)
"	C1	41	(2)
"	C2	42	(2)
Transverse	D8	16½	Untreated
"	D9	21½	"
"	C11	54	(1)
"	C12	52	(1)
"	C13	54	(1)
"	C14	47	(1)
"	C9	20½	(2)
"	C10	14	(2)

SUBMARINE ENGINES.

By A. P. Chalkley, B.Sc., A. M. Inst. C. E.

Whilst it is true that the development of the submarine in its present form would have been almost an impossibility had it not been for the internal-combustion engine, it is even more apparent that the capacity of the later type to play an important rôle in offensive warfare is due almost entirely to the great progress that has been made during recent years in the construction of the Diesel oil engine. It is true that in certain cases (for instance, in some of the French submersibles) steam turbines are employed for propulsion on the surface, but these exceptions are not numerous and general opinion is unfavorable towards such craft. At any rate, they do not appear to have been so successful in practice as the boats in which the surface propelling machinery consists of oil engines.

All the earlier submarines built in this country and in Germany, as well as most of those in France, were equipped with petrol or paraffin engines, but it was very soon apparent that the employment of such motors put a very distinct limitation upon the possibilities of the submarine. Quite apart from the danger involved, which was by no means negligible, the construction of petrol motors of high power was not to be considered for this work, and above all, owing to the relatively high fuel consumption, the radius of action of a petrol-engined submarine was too small to enable the boat to take part in extensive operations, even were it suitable in other ways. Yet, full credit should be given to the petrol-engine manufacturers who ten years ago undertook to construct engines of 200 or 300 H.P. and built them so well that, even now, several of them are still doing very good work. In most of the early British craft a horizontal opposed type was employed, but in many of the foreign submarines vertical motors were installed, and Fig. 1 shows such a machine built by Messrs. Thornycroft for the Italian Navy about 1905 or 1906.

The first Diesel-engined submarines were completed seven or eight years ago, and there was very little difference between the times when the earliest British, German and French craft with the new propelling machinery were commissioned. Probably the idea originated in France, but the earlier boats were about four years under construction, so that it was, if anything, later before they were put into service. Far more difficulties were encountered in the manufacture of these motors than was anticipated, the troubles being mainly due to the high speed of rotation which was necessary in order to reduce the weight of the machinery to a minimum and to diminish the size in view of the limited space available for the installation. The construction of the air compressors for the injection of the fuel was in itself no mean problem, and it was also very difficult for the designers to overcome the troubles which arose in consequence of the high temperature of combustion. This may be in the neighborhood of 1,200 to 1,500 degrees centigrade, and as the number of explosions per unit of time is much greater than in a slow-running engine, the heat problem is rendered more difficult of solution. It is not surprising, therefore, that in the earlier submarine engines, there were many accidents owing to inefficient piston cooling, and the employment of steel pistons (with the object of gaining greater strength) working in cast-iron cylinders—a method instituted by one or two French builders—did not improve matters and was quickly abandoned.

It is only necessary to quote the respective weight of slow-running Diesel engines as employed on mercantile vessels, and the high-speed sets used on submarines to show the vast difference there must be in the construction of the two types. In the first case the weight, including accessories, is generally in the neighborhood of 350 pounds per B.H.P., whilst submarine engines on an average do not weigh more than 50 to

70 pounds per B.H.P., usually the lower figure. It need hardly be pointed out that the greatest care has to be exercised in weight cutting in Diesel engines, as the excessive pressures that have to be withstood do not allow of any weakening of the main structural parts. It is for this reason that most of the German submarine engines are built largely of manganese bronze, which, although adding somewhat to the expense, is thought by the German designers to be the most satisfactory method.

The requirements of submarine engines which have just been mentioned—smallness and light weight—would naturally lead to the conclusion that the two-cycle motor should offer great advantages for the work. This is to a certain extent the case for a Diesel engine operating on the two-cycle principle should, apart from exceptional conditions, occupy less space and have a lower weight than the four-cycle engine developing the same power. In practice, however, the difference does not appear to be very large, and many engineers are of opinion that the greater difficulties involved in the construction of the two-cycle type do not warrant its adoption for the relatively small advantages gained.

It is a matter of particular interest that in Germany, although the four-cycle Augsburg motor was originally employed, the two-cycle Diesel engine is now invariably adopted for all submarines, this policy having been followed for some years past. In England, on the other hand, nearly all the machinery for the Diesel-engined submarines has been built by Vickers, who have employed the four-cycle principle, although their motors are distinctive in that they avoid the use of the air compressor for injection, since solid or mechanical injection of fuel is utilized. This naturally effects a saving in weight and space, and although the Vickers engines of 850 H.P. in the modern British submarines are eight-cylinder four-cycle machines, and some of the German submarine motors of about the same power have only six working cylinders and operate on the two-cycle principle, there is probably very little difference either in the weight or in the overall dimensions of the engine. This is not difficult to explain when it is considered that there are no injection-air pumps on the Vickers motors, whereas in the Krupp engines referred to there are two scavenger pumps and two air compressors. For some of the new British submarines, however, two-cycle engines have been constructed.

In France both two and four-cycle engines have been installed, some of each type having been built in Germany for the purpose, besides those constructed in France, but it is somewhat curious that in Sweden, where the Polar Co., of Stockholm, have in the past year or two taken up the manufacture of submarine engines, the four-cycle type has been adopted in spite of the fact that the success of this firm has been built up on the manufacture of motors of the two-cycle principle. In Switzerland Messrs. Sulzer build two-cycle submarine engines, whilst in Italy a similar type manufactured by the Fiat Co. has been adopted in recently constructed submarines.

It would thus appear that, on the whole, the two-cycle engine has found most general acceptance and when the submarine develops into a much larger and faster craft than it is at present, it appears probable that the two-cycle type will have to be employed, owing to its general suitability for higher powers. Owing to the complications involved in making them directly reversible, most four-cycle submarine engines have hitherto been constructed as non-reversible sets, the electric motors used for under-water propulsion being employed for reversing. Exception to this should be made in the case of the Polar engine, which will be referred to later, but it may be mentioned, on the contrary, that practically all the two-cycle engines are directly reversible, and it can easily be understood that the design of a two-cycle reversible motor is much simpler than that of the four-cycle type unless some special means be adopted.

The differences in design of the various types of Diesel engines em-

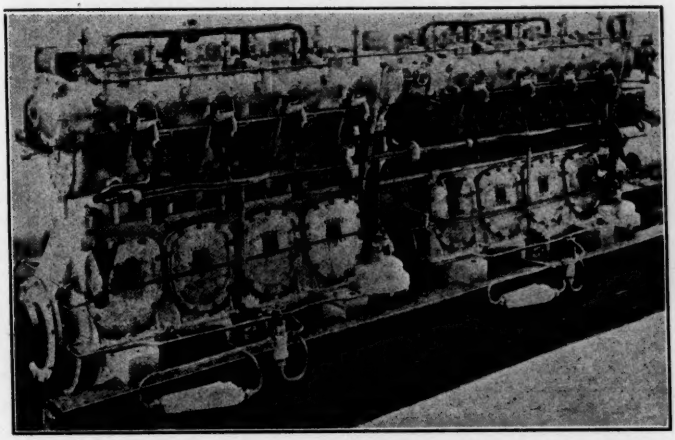


FIG. 1.—A THORNYCROFT SUBMARINE ENGINE BUILT EIGHT YEARS AGO FOR THE ITALIAN NAVY.

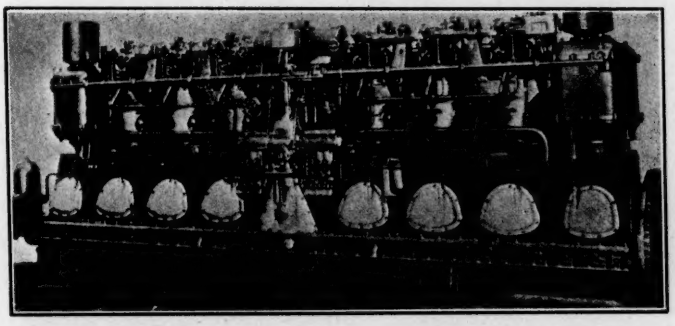


FIG. 2.—KRUPP SUBMARINE DIESEL ENGINE OF 900 B.H.P. FOR THE GERMAN NAVY.

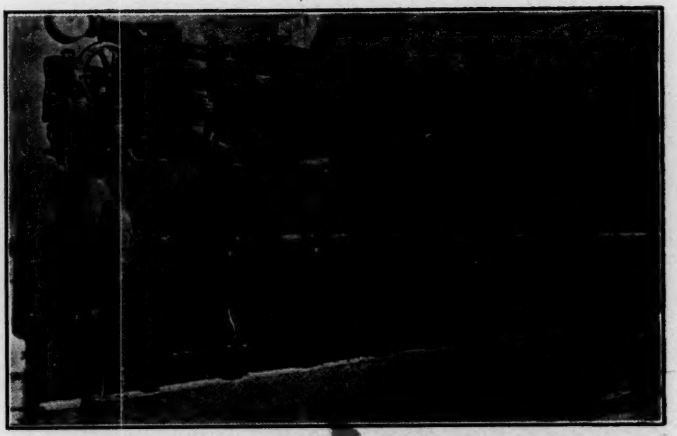


FIG. 3.—A 900 B.H.P. NURNBERG SUBMARINE MOTOR.

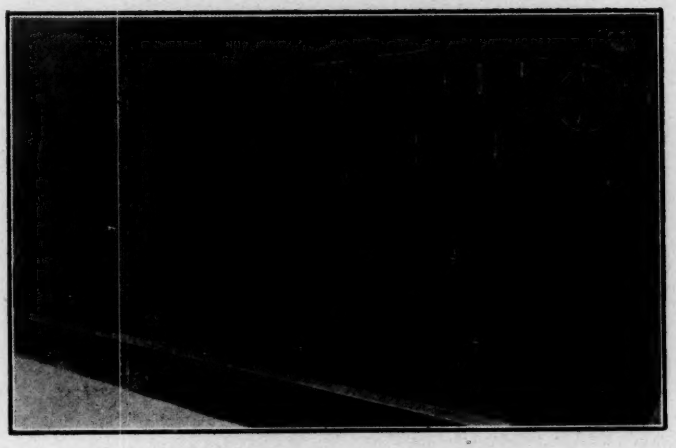


FIG. 4.—A TWO-CYCLE NURNBERG SUBMARINE MOTOR.

ployed on submarines as built by the different manufacturers are quite remarkable, and the fact that practically all of them have done excellent work is a clear indication of the diverse manners in which a problem, with one end in view, can be worked out successfully. Nothing further need be said regarding the motors used on British submarines, but some details may be given of the types employed in the underwater craft of other Navies. In Germany, Krupps, of Kiel, and the Maschinenfabrik Augsburg-Nürnberg, build all the motors used in German submarines, and the former type is generally considered to be the more successful, although reports are somewhat at variance.

Fig. 2 shows one of the standard Krupp engine which is installed in many of the very latest German submarines, from U17 onwards, being designed to develop about 900 B.H.P. at 450 r.p.m. It is a two-cycle six-cylinder machine with two air compressors driven off the crankshaft, between the working cylinders, and two scavenger pumps also directly coupled to the crankshaft and arranged at the extreme ends of the engine. The great difficulty in the design of these high-speed two-cycle engines is to ensure an effective supply of scavenging air to the cylinder, and to cope with this trouble there are three scavenging valves in each cylinder cover operated from one cam on the camshaft by means of a linkwork. As these valves occupy practically the whole available space in the cover, the fuel-inlet and starting valves have to be set diagonally and the arrangement can be seen from the illustration. The engine is directly reversible and is, of course, practically enclosed, as are all the high-speed submarine Diesel motors.

The Nürnberg engine is in some ways more interesting than the Krupp machine. It is also designed for an output of about 900 B.H.P. at 450 r.p.m. and operates on the two-cycle principle. It has eight cylinders, each 310 mm. in diameter and 340 mm. stroke, and the two two-stage air compressors are driven directly off the end of the crankshaft. Figs. 3 and 4 show two opposite engines complete for installation in a submarine, and it will be noticed that the wheels and levers for controlling the engines are opposite each other when the motors are actually fitted in the vessel.

The chief point in the design of this engine (which is well illustrated in Figs. 5 and 6, the former being a sectional front elevation, and the latter the end elevations) is the fact that the scavenging pumps are arranged as stepped pistons immediately below the working cylinders, so that there are in all eight such pumps. It might be thought that this would give an excessive supply of air, but it must be remembered that the actual scavenging volume in each pump cylinder is not large, corresponding only to the difference between the diameters of the working and scavenge cylinders. The actual diameter is 475 mm., and the stroke of the pump is, of course, the same as that of the working piston, viz: 340 mm. The high-pressure cylinder of the compressors has a diameter of 100 mm., and the low-pressure 300 mm., the stroke for both being 250 mm. The overall length of this engine is under 23 feet, and the height is about 8 feet 6 inches.

The piston-cooling arrangement in the motor is somewhat unusual, and is carried out by pumping oil at a pressure of about 35 pounds per square inch through the crankshaft, and connecting rod, and thence to the gudgeon pin, after which it passes through a pipe to the body of the piston. It is then returned to the oil cooler and used over again; this method is interesting in contrast to the one employed on the previously-described motor, where water cooling is adopted. There is only one scavenging valve to each cylinder, and this is inclined to the axis of the cylinder, as is also the fuel-inlet valve on the opposite side. The method of reversing adopted in this case is to move the cam shaft through an angle of 30 degrees, which sets the scavenge valve and fuel valve at

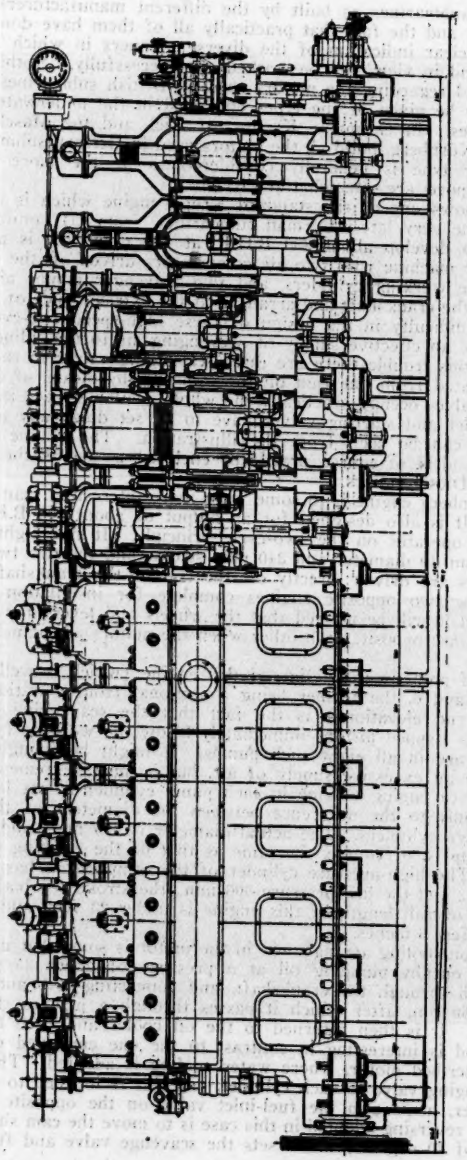


FIG. 5.—FRONT ELEVATION AND SECTION OF THE 900 B.H.P. NURNBERG ENGINE SHOWN IN FIG. 4.

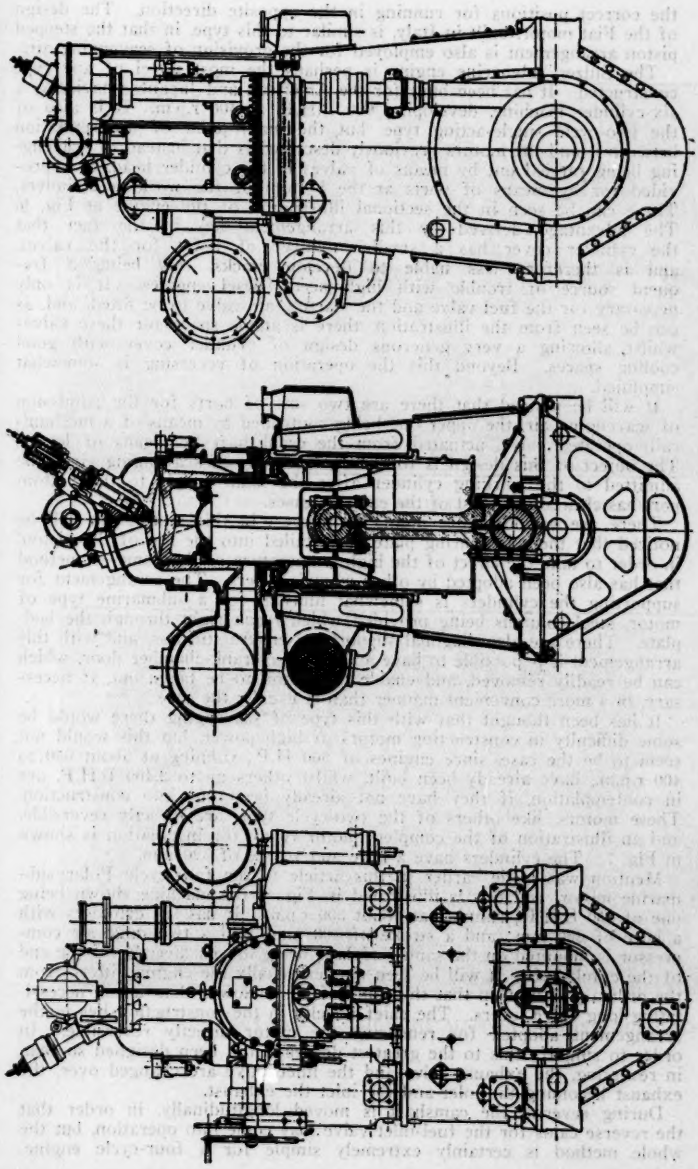


FIG. 6.—END ELEVATIONS AND SECTION OF A PAIR OF NURNBERG TWO-CYCLE DIESEL ENGINES.

the correct positions for running in the opposite direction. The design of the Fiat motor, built in Italy, is similar to this type, in that the stepped piston arrangement is also employed for the provision of scavenging air.

The Sulzer submarine engine is perhaps the most novel of any type constructed. It has been built for the American and Japanese navies as a six-cylinder machine, developing 600 B.H.P. at 400 r.p.m. It is also of the two-cycle single-acting type, but the main point of differentiation between it and the motors previously described is that instead of scavenging being carried out by means of valves in the cylinder head, it is provided for by means of ports at the bottom of the working cylinders. These can be seen in the sectional illustration of the engine at Fig. 9. The advantage derived by this arrangement lies in the fact that the cylinder cover has a smaller number of holes for the valves, and is therefore less liable to develop cracks, this being a frequent source of trouble with high-speed Diesel engines. It is only necessary for the fuel valve and the starting-air valve to be fitted, and, as can be seen from the illustration, there is ample space for these valves whilst allowing a very generous design of cylinder cover with good cooling spaces. Beyond this the operation of reversing is somewhat simplified.

It will be noticed that there are two sets of ports for the admission of scavenging air, the upper one being controlled by means of a mechanically-operated valve, actuated from the crankshaft by means of levers. The object of this design is for an extra supply of scavenging air to be admitted to the working cylinder after the main supply to the bottom port has cleared out most of the exhaust gases.

There are various other points of interest in this motor. It will be noticed that there is a firing plate, dove-tailed into the top of the piston, in order to take the effect of the high temperature of the flame—a method that has also been adopted by other manufacturers. The arrangement for supporting the cylinders is somewhat unusual for a submarine type of motor, steel columns being provided which reach right through the bedplate. There are also diagonal supports to ensure stiffness, and with this arrangement it is possible to have a very large crank-chamber door, which can be readily removed, and enable the piston to be taken out, if necessary, in a more convenient manner than is usually the case.

It has been thought that with this type of scavenging there would be some difficulty in constructing motors of high power, but this would not seem to be the case, since engines of 900 H.P., running at about 350 to 400 r.p.m., have already been built, whilst others up to 2,400 B.H.P. are in contemplation, if they have not already been put into construction. These motors, like others of the two-cycle type, are directly reversible, and an illustration of the complete motor ready for installation is shown in Fig. 7. The cylinders have a bore and stroke of 320 mm.

Mention was made earlier in this article to the four-cycle Polar submarine motor, and this is illustrated in Fig. 8, the machine shown being one of 350 B.H.P., running at about 500 r.p.m. It has six cylinders with a bore of 290 mm., and a stroke of 300 mm., and a two-stage air compressor is mounted on the same bedplate, being driven directly off the end of the crankshaft. It will be seen that externally the engine differs from the ordinary design, in that the camshaft is set lower than usual, necessitating long valve levers. The chief novelty in the construction lies in the arrangement adopted for rendering the motor directly reversible. In order to simplify this to the greatest degree it has been designed so that, in reversing, the exhaust valve and the inlet valve are changed over, the exhaust becoming the inlet and the inlet the exhaust.

During reversal the camshaft is moved longitudinally, in order that the reverse cams for the fuel-inlet valve may come into operation, but the whole method is certainly extremely simple for a four-cycle engine.

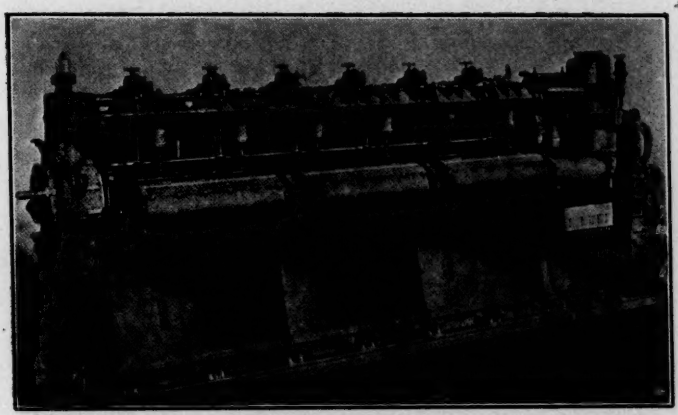


FIG. 7.—A SULZER TWO-CYCLE ENGINE OF THE TYPE USED IN THE AMERICAN AND JAPANESE NAVIES.

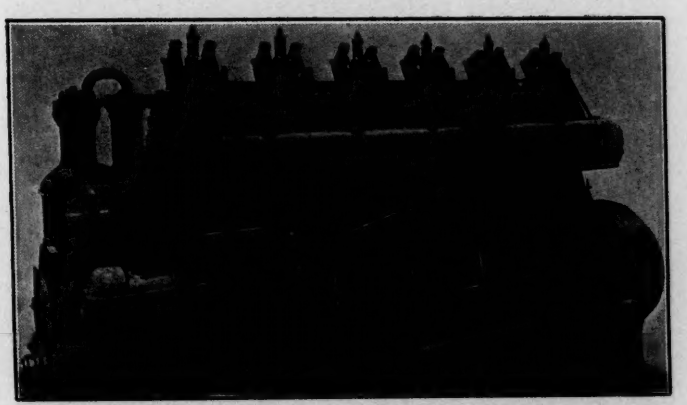


FIG. 8.—A POLAR FOUR-CYCLE ENGINE INSTALLED IN SWEDISH SUBMARINES.

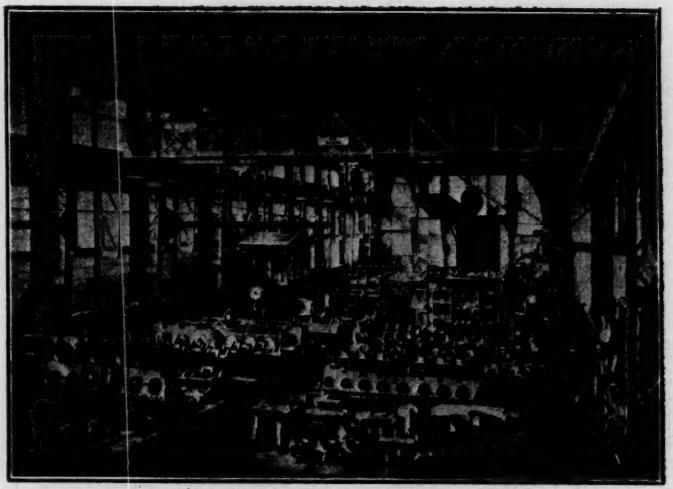


FIG. 10.—KRUPP'S SUBMARINE-ENGINE ERECTING SHOP.

FIG. 11.—SUBMARINE ENGINES BEING TESTED IN THE SHOPS AT KRUPP'S WORKS.

Motors of this type are now built in fairly large sizes, at any rate, up to about 600 B.H.P.

It is no secret that, even before the war, all the leading naval powers were endeavoring to produce a submarine of much larger size, and with a higher speed than the types now being used, and the present war has, of course, only caused the naval authorities to hasten towards this end. The development of the submarine is only limited by the limitations of the oil engine, and immediately satisfactory motors of 2,000 or 3,000 B.H.P. can be produced, submarines of much larger size and with speeds of 20 knots or over will be constructed. The difficulties in the manufacture of such high-powered motors running at speeds of 400 r.p.m. are enormous, and a considerable period may elapse before complete success is attained.

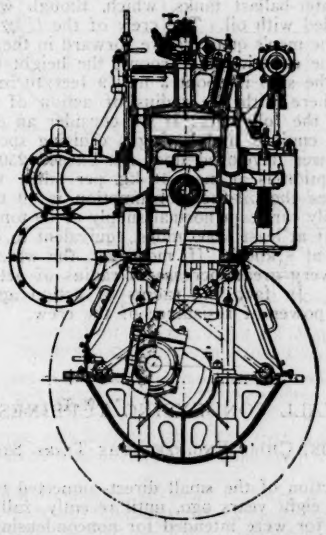


FIG. 9.—SECTIONAL END ELEVATION OF A SULZER TWO-CYCLE ENGINE.

It must not be thought, however, that it is only the large type of submarines that is required in the future, for it is evident that the purely defensive submarine will figure very largely in naval programs. This is particularly the case in the smaller countries whose navies cannot hope to cope with those of the larger powers, but who may be able to maintain a very strong defensive by the employment of numerous small submarines. It is for this reason that a very great development of the submarine-engine industry may be expected in the next few years, for although the submarine may not, as has been suggested in some quarters, render the battleship obsolete, yet it is bound to play a very important part in all naval questions in the next 10 or 15 years.—"Cassier's Engineering Monthly."

GERMAN SUBMARINES.

The Italian naval review "Rivista Marittima" gives in its issue for March an abstract of an article written by a German officer, according to which the German submarine *U 47* completed her trials in February last. The number of submarines in the service at the opening of hostilities was 27. Adding those then in course of construction for Germany, the number reached 36. New units have been put down since the war was declared, besides those then in course of construction for foreign governments. The crews for submarines, when not in action, are quartered in hulks which are anchored in German waters. When a submarine leaves for a cruise she is so filled with stores and fresh water that only a very narrow passage remains free. Naphtha exudes at the steel bulkheads, doors, and throughout the whole ship, this lending confirmation to the rumor that the water-ballast tanks, which, though watertight, are not oiltight, are also filled with oil. The crew of the *U 47* and similar units number 30 men. The men's quarters are forward in the torpedo-launching room, and aft in the electric-motor room; the height of these rooms at the center line of the ship is about 3 m. (9 feet 10 inches). The same review deals elsewhere with the radius of action of the German submarines, and states the following: If we consider an 800-ton submarine, propelled by Diesel engines, and having a cruising speed of 8 knots on the surface, the power developed does not exceed 250 horsepower, and the oil-fuel consumption is at most 10 kg. per mile. On this basis, for every thousand miles the oil-fuel consumption is 10 tons. Considering that the boat probably carries a normal supply of 50 tons, she would have a radius of action of at least 5,000 miles, equivalent to over 600 hours of constant navigation at 8 knots. If, moreover, the water-ballast tanks be filled with oil, the very great increase in radius of action thereby made possible is obvious. In fact, the radius of action appears now to be limited only by the power of resistance of the crew.

SMALL CONDENSING TURBINES.

By W. J. A. London, Chief Engineer, the Terry Steam Turbine Co.

Since the introduction of the small direct-connected turbine on a commercial basis, some eight years ago, until recently, fully 90 per cent. of the machines called for were intended for noncondensing service. In the few cases where they were called upon to operate condensing, such as for marine work, little attempt was made at economy, as the operation of these machines condensing was more a matter of convenience than of water rate. It has been acknowledged that the designing of small turbines is much different from that of large machines, for were a small turbine designed on the same principles and lines as a big machine a hopeless commercial failure would result. There have, therefore, been two distinct fields in turbine work; the principles governing the designs of small and large machines being so much at variance that they might be said to be almost as different as the designs of a steam and a gas engine.

Most small turbines were installed to operate noncondensing, being used primarily for auxiliary apparatus, the exhaust being used in feed-water heaters. Small isolated plants were operated noncondensing, the exhaust steam being used for industrial or heating purposes.

For isolated plants, such as small pumping installations of, say 150 to 500 H.P., where economy was of much importance and the saving by

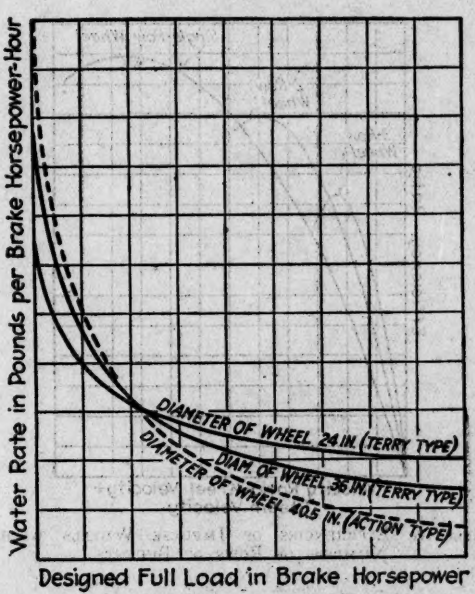

Water Rate in Pounds per Brake Horsepower-Hour
DIAMETER OF WHEEL 24 IN. (TERRY TYPE)
DIAM. OF WHEEL 36 IN. (TERRY TYPE)
DIAMETER OF WHEEL 40.5 IN. (ACTION TYPE)
Designed Full Load in Brake Horsepower

FIG. 1.—RELATIVE EFFECTS ON WATER RATE OF VARYING DIAMETER OF WHEEL AND DESIGNED FULL LOAD.

The characteristics shown are due to increased windage losses on the larger wheels. All curves are plotted for the same conditions of steam pressure.

operating condensing sufficient to offset the first cost, maintenance, etc., the turbine has been at a disadvantage as compared with the reciprocating engine.

The average thermal efficiency of small turbines is in the neighborhood of 40 per cent. With this efficiency all exhaust steam can be utilized without "blowing off," so that no higher efficiency is required or even desirable.

Forty per cent. efficiency does not represent the highest available in this class of machine, but it is conceded to be about the highest commercial efficiency. The cost of small turbines varies approximately directly as the square of the diameter of the runner, whereas the efficiency is increased approximately only inversely as the square root of the runner diameter, so any saving in steam consumption must be accompanied by a marked increase in first cost.

Above 500 to 600 H.P. the field of the large turbine, where water-rate efficiency is of paramount importance, is approached. These machines operate condensing in the same proportion that the small machines operate noncondensing, and the whole problem of design must be attacked on a totally different fundamental basis.

For powers of, say 150 to 500 H.P., condensing, where high efficiencies were desirable, the reciprocating engine until recently had no serious competitor in the steam turbine. The reason for this is obvious. The reciprocating engine was developed, and the meager demand for small

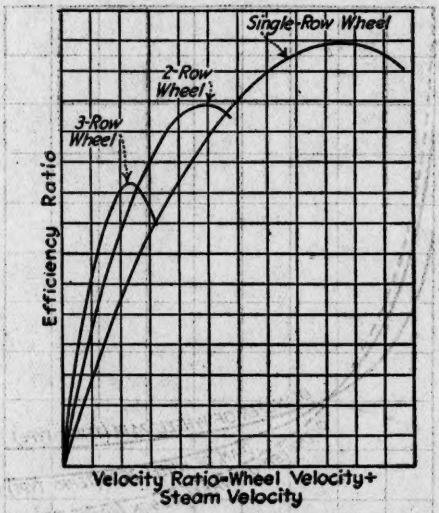


FIG. 2.—RELATIVE EFFICIENCIES OF IMPULSE WHEELS WITH VARIOUS NUMBERS OF ROWS OF BUCKETS.

high-efficiency turbines did not warrant the manufacture of special machines, and furthermore, the customer would not pay the price that would have to be charged.

Conditions have changed rapidly during the last two years or so, and there is now a big demand for small condensing turbines of high efficiency, both high- and low-pressure, which has led to the development of a third class of machine to meet the requirements of this market. To distinguish this class from the small and the large machines, it is permissible to call it the "intermediate design." This design should have, as far as possible, the simplicity and accessibility of the small machine with an efficiency approaching that obtainable with the larger units.

When Parsons and DeLaval built their first turbines the main trouble was not with the turbine itself, but with the "other end," or the driven unit. A turbine is of little use by itself, and it was not until it was demonstrated that it had come to stay that generator, pump and blower makers awoke to the fact that they must remodel their apparatus to meet turbine requirements.

Rapid as the turbine development has been, it would have been more so had it not been for the slow development of the "other end." And past events have again repeated themselves in the field of the "intermediate design." This machine would not have been possible had it not been for the rapid strides that have taken place along the following lines: (a) The manufacture at reasonable cost of high-tensile steel for turbine wheels; (b) the increase in permissible speed of generators, blowers, etc.; and (c) the introduction on a commercial basis of the speed-reducing gear.

In "Power" of October 28, 1913, a brief description was given of the return-flow condensing steam turbine that had just been developed by the Terry Steam Turbine Co. for this so-called "intermediate field."

On the completion of tests of the first machine, several modifications and improvements naturally suggested themselves and are incorporated in the latest designs. For turbines of small power the latest tests show some remarkably good efficiencies, as will be seen by the details of the tests published herewith.

One of the main changes in design has been to carry the principle of the horizontally-divided case to the last extreme. In the original return-flow machine the casing was divided horizontally, but the center diaphragms and the center-diaphragm glands were not, whereas in the machine shown all diaphragms and diaphragm and end glands are thus split, reducing disassembling and assembling time to a minimum. See Fig. 5, right.

In the larger frames another important change has been made. The high-pressure wheel of the Terry type which was incorporated in the first machine has been superseded by a two-row multi-velocity type of wheel running at a high peripheral speed. Extensive experiments have shown that, up to certain peripheral speeds and certain powers with a given thermal drop, the Terry type of bucket is well adapted, but beyond this range, the two-row bladed wheel has the advantage. See Fig. 5.

There are several factors entering into the correct design of a wheel of this type other than the actual or theoretical blade efficiency, which make this problem interesting and more complex than one would suppose from a superficial study of the subject on a purely blade-efficiency basis. Disc friction, the power transmitted or rated power of the machine, commercial considerations regarding first cost (which controls the selling price), are all big factors independent of any blade-efficiency theory.

The return-flow turbine is designed so that the pressure in the first stage will be about 2 to 5 pounds above the atmosphere. With ordinary steam pressures of, say 150 pounds, and allowing two impulses, the peripheral velocity of the buckets must be about 636 feet per second for best efficiency. At 3,600 r.p.m. this calls for a pitch diameter of 40½ inches. For three reversals the diameter of wheel would be in the neighborhood of 24 to 26 inches.

Fig. 1 shows the relative efficiencies of three types of wheels and the important relation that skin friction and windage bear to the overall efficiency. A "two-velocity stage" wheel is more efficient from a blade-efficiency standpoint than a "three- or four-velocity stage" (see Fig. 2), yet the friction created by the increased diameter is far more undesirable than one would at first imagine, and the advantage gained by augmented blade efficiency is more than offset by the added losses in other directions.

The various formulas of Stodola, Lewicki, Odell and others, for skin friction of discs, show clearly how big a factor this can be, and while these formulas are somewhat vague and indefinite regarding certain conditions that have to be taken into account, they all agree that this friction loss varies as the second to the 2.5 power of the diameter of the wheel and nearly as the cube of the peripheral velocity. Again, these formulas do not take into account the windage of the blading, which is obviously greater in a bladed wheel than on a bucket wheel.

In reverting to the bladed type of wheel in the high-pressure end, instead of the semicircular-type bucket, it is interesting to note in passing that the first turbine experimented with by E. C. Terry in 1893 was on this principle. Fig. 3 shows the wheels and guide blades of this early machine, the patent number of which is 508,190.

The Terry type of machine is, primarily and essentially a noncondensing turbine. Its simplicity and consequent unlikelihood of derangement make it an ideal machine for the duties that it is called upon to perform. Within certain limits of speed, vacuum, etc., the two- or three-stage Terry combination makes an equally good condensing machine, having a thermal efficiency as high as that of the single-stage noncondensing design; but

FIG. 3.—WHEELS AND GUIDE VANES OF EARLY TERRY TURBINE.

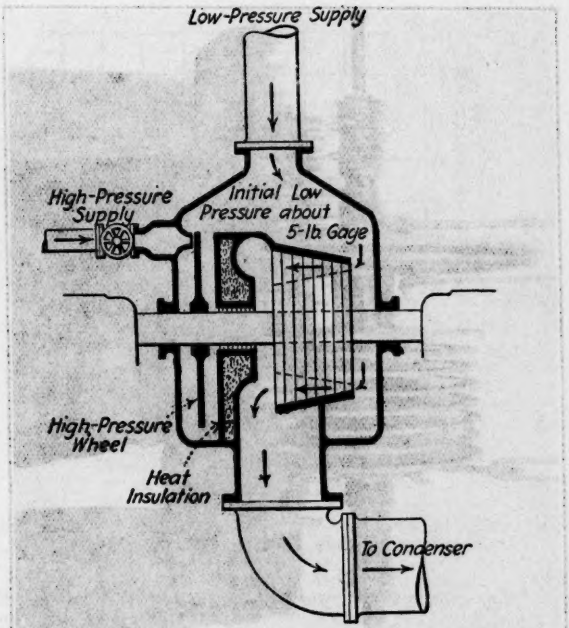


FIG. 4.—DIAGRAMMATIC SECTION OF RETURN-FLOW TURBINE.

when confronted with the necessity for high speeds, high vacuum and larger powers, the wheel becomes impracticable at the low-pressure end owing to its inability to handle a large volume of steam to the best advantage. In the later machines, therefore, the low-pressure wheel has been replaced by a series of single velocity-stage impulse wheels. That practically all authorities agree that this type of wheel for low-pressure work forms the ultimate turbine element is evidenced by the fact that it is being adopted by practically all turbine builders of both large and small machines, with the one exception of the builders of the reaction, or Parsons, type; and that this type of machine is not adaptable to small powers is evidenced by the fact that the builders themselves resort to the impulse principle in their smaller designs.

Again, the "composite design" of velocity staging in the high-pressure end and pressure staging in the low-pressure must be the last word in turbine development if latest designs of practically all the turbine builders both here and in Europe are any criterion.

The obvious advantage of high vacuum in a turbine, particularly in a low-pressure turbine, with the difficulty of designing, building or keeping glands vacuum tight without the necessity of a water seal with its attendant piping and subsequent sediment troubles, led to the departure from the orthodox straight-flow principle to the return-flow design for the elimination of this long-standing bugbear in turbine work. This question of gland leakage often results in trouble between the turbine and the condenser makers when trying to meet guarantees, while the customer

FIG. 5.—ROTORS OF OLD AND NEW DESIGNS OF RETURN-FLOW TURBINES.

Old design of rotor having Terry high-pressure bucket wheel. Rotor of new machine having two rows of impulse blading on the high-pressure wheel.

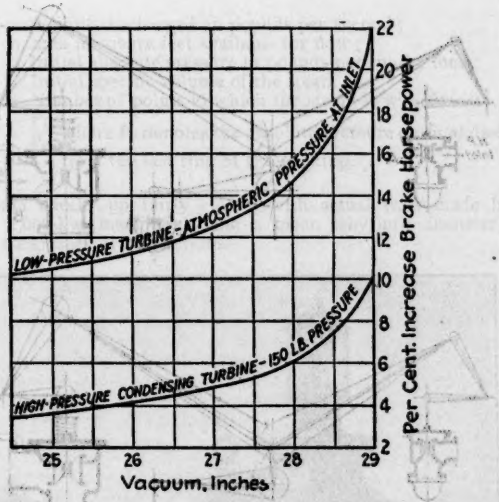


FIG. 6.—PERCENTAGE INCREASE IN POWER AVAILABLE (THEORETICAL) PER INCH VACUUM WITH CONDENSING TURBINES.

looks on and sees the machine run at a lower vacuum than called for and pays the coal bill anyway.

It is often advocated that with a steam seal on the glands and a little steam blowing outward into the engine room there cannot be any air leaking into the turbine. This contention is wrong, as has been demonstrated many times. It often happens that there is a counter current going on, air traveling along one part of the gland and steam passing out of the glands in the opposite direction, this condition being the hardest possible phenomenon to detect. Fig. 4 shows diagrammatically the construction of the return-flow turbine with the low-pressure end in a complete envelope of steam above atmospheric pressure, eliminating the possibility of air leaking into the casing. The rotor and the lower half of the casing of the return-flow machine are shown in Fig. 5.

One other important feature in connection with the arrangement of glands on the return-flow turbine is that no supplementary steam supply is necessary for sealing them when under full load, and even at light loads any steam that finds its way through must pass through the low-pressure end of the turbine, thereby doing work, whereas with the ordinary type of steam-sealed glands all the steam which does manage to escape goes directly to the condenser without doing any further work. That this auxiliary steam supply can amount to quite a factor is evidenced by various tests that have been made. Of course, when a machine is new the glands are tight, so that the leakage during this period is imperceptible, but after setting the machine for commercial operation or if it has been in operation for some time, it is hard to know without repeated tests what this steam leakage amounts to. In big machines this is never, however, a serious amount, but in small ones it can be quite a big percentage of the total steam used.

A careful analytical study of the performance of labyrinth glands was

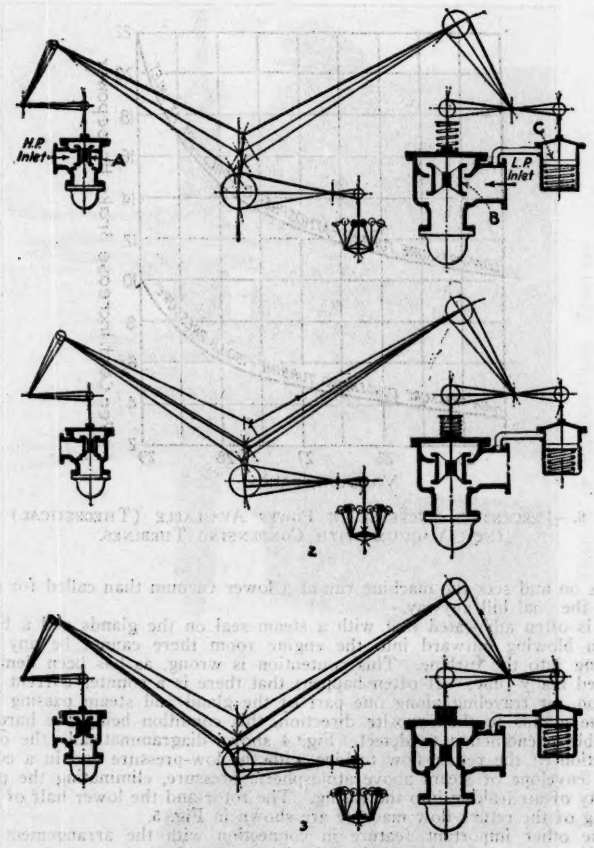


FIG. 7.—DIAGRAMS OF LINK MOTION TERRY MIXED-PRESSURE GOVERNOR CONTROL (RATEAU SYSTEM).

1. Position when starting turbine; high and low-pressure valves open. Piston C forced down by ample supply of low-pressure steam.
2. Speed regulation: High-pressure valve closed. Turbine running on ample supply of low-pressure steam.
3. Pressure regulation: Low-pressure supply stopped; piston lifted by spring, closing low-pressure valve and opening high-pressure valve. Governor is always free to close both valves if load is suddenly taken off the turbine.

made and published by H. M. Martin, and the formula derived from his experiments is given in his book on steam turbines, as follows:

$$W = 68\,A\sqrt{\frac{P_1\left(1-\frac{l}{x^2}\right)}{V_1\,(N+\log_e x)}}$$

where

W = Weight discharged in pounds per second;
A = Area in square feet available for flow;
P_1 = Initial absolute pressure in pounds per square inch;
V_1 = Initial specific volume of the steam;
N = Number of points at which the steam is wire drawn;
$x = \frac{P_1}{P_2}$, where P_2 denotes the absolute pressure on final discharge from the last ring of the packing.

This formula checks up fairly closely with actual tests made by the writer on a 2,000-kw. machine having a mean labyrinth diameter of 8 inches and 12 elements or restrictions.

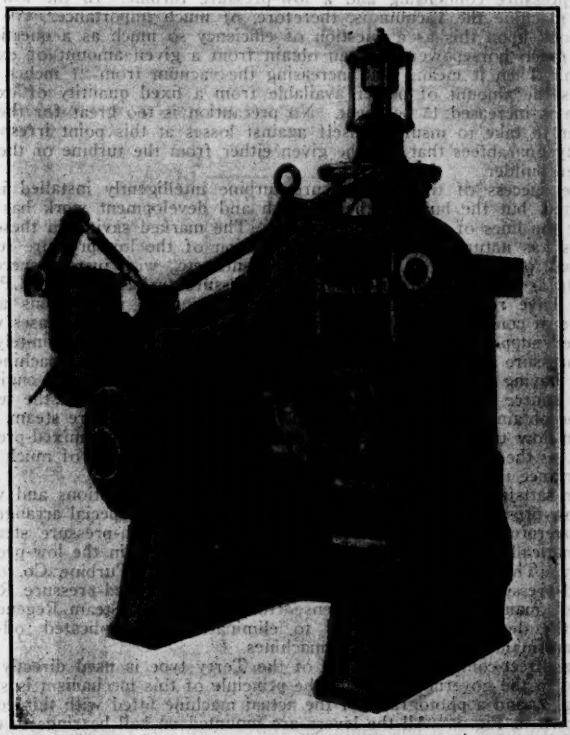


FIG. 8.—TERRY RETURN-FLOW TURBINE WITH RATEAU MIXED-PRESSURE CONTROL MECHANISM.

From the formula it will be seen that the amount of steam passed by a labyrinth gland is directly proportional to the diameter of the glands. This diameter of gland does not follow any relation to the output of the turbine, and it will be seen that the smaller the machine the larger the

percentage of steam that will be passed by the gland; so, as mentioned above, while the amount of steam passed by a labyrinth gland in large machines can be an insignificant factor, it is obvious that in the small machines it can be a serious item. For instance, the figures mentioned above in connection with the 2,000-kw. machine show the total steam passed as 177 pounds per hour. On the basis of 15 pounds per kw. this would give a percentage loss due to the glands of 0.6 per cent., whereas reducing this quantity in the ratio of the diameter of the glands, namely, 8 inches to, say 5 inches on a 200-kw. machine, the gland leakage would be reduced to 111 pounds, but the percentage of the total steam consumption would be increased to 2.5 per cent., the latter based on a water rate for the smaller machines of 22 pounds per kw.-hour.

Fig. 6 shows the theoretical saving per inch of vacuum in a straight high-pressure condensing and a low-pressure turbine. In the low-pressure machine the vacuum is, therefore, of much importance. We must not look upon this as a question of efficiency so much as a question of how much horsepower one can obtain from a given amount of exhaust steam. Then it means that increasing the vacuum from 27 inches to 28 inches the amount of power available from a fixed quantity of exhaust steam is increased 15 per cent. No precaution is too great for the purchaser to take to insure himself against losses at this point irrespective of any guarantees that may be given either from the turbine or the condenser builder.

The success of the low-pressure turbine intelligently installed is undisputed, but the bulk of the research and development work has been along the lines of the larger machines. The marked saving in these mechines has naturally led to the introduction of the low-pressure turbine in small plants such as breweries, ice plants, etc., with just as successful results as with the larger units. Low-pressure turbines of 50 H.P. and more have recently been installed, and many more installations are in course of construction. With the exception of a few isolated cases where a fixed supply of exhaust steam can be depended upon indefinitely, the low-pressure turbine has given place to the mixed-pressure machine, the latter having the advantage that should anything happen to the engine or other source of low-pressure steam supply, the full power of the turbine can be obtained when operated with high- or mixed-pressure steam. The return-flow turbine is particularly applicable to low- and mixed-pressure work, as the effect of vacuum in a machine of this kind is of much more importance than in a high-pressure condensing machine.

For satisfactory operation under mixed-pressure conditions and where the low-pressure supply is liable to decrease or fail, a special arrangement of governor mechanism is designed, so that the high-pressure steam is automatically admitted to make up for any deficiency in the low-pressure supply. The system employed by the Terry Steam Turbine Co. on its mixed-pressure turbines is what is known as the mixed-pressure Rateau control, manufactured under license from the Rateau Steam Regenerator Co., the design being modified to eliminate the complicated oil-relay mechanism necessary on larger machines.

The direct-connected governor of the Terry type is used directly connected to the governor valves. The principle of this mechanism is shown in Fig. 7, and a photograph of the actual machine fitted with this control is shown in Fig. 8. All the levers are mounted on ball bearings to eliminate friction as far as possible. The governor running at high speed has more power than the usual low-speed geared governor and the mechanism itself being balanced by counterweights as shown in Fig. 8, the governor is relieved of all external loads other than to operate the balanced valves.

In all turbine practice, both in high- and low-pressure machines, the question of pipe connections and the elimination of stresses on the turbine is well known to be a serious problem. This is particularly so with the

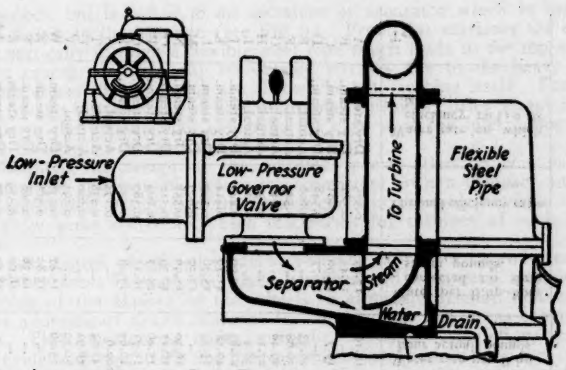


FIG. 9.—ARRANGEMENT OF LOW-PRESSURE STEAM INLET ON RETURN-FLOW TURBINE.

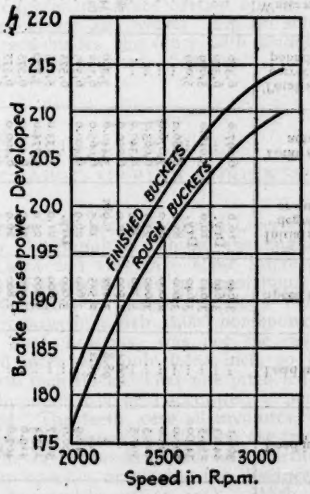


FIG. 10.—TESTS OF IMPULSE WHEELS, SHOWING EFFECT OF FINISHING BLADES.

big piping necessary for low-pressure machines, and it is again of further importance in the smaller units, which, on account of their compact size and light weight are susceptible to distortion from outside stresses. The common practice is to bolt the low-pressure valve and piping directly to the turbine casing. This entails considerable risk of pulling the turbine and, consequently, the whole unit out of line, causing vibrations and generally unsatisfactory running. To eliminate this in the return-flow turbine the low-pressure steam supply is not rigidly connected with the tur-

PERFORMANCES OF TERRY RETURN-FLOW TURBINES.

Turbine number.	Initial steam pressure before governing valves, pounds.	Vacuum exhaust, ins., Hg. referred to 30-inch barometer.	Load in B.hp.	Load in kw.	Speed in r.p.m.	Initial steam condition, degrees F. superheat or quality.	Total steam per hour, actual, pounds.	Total steam per hour corrected to dry steam, pounds.	Water rate per kw.-hour actual pounds.	Water rate per B.hp.-hour actual, pounds.	Water per B.hp.-hour corrected to dry steam, pounds.	Thermal efficiency ratio	Water rate on which efficiency ratio is based.	Test.
1883	128.8	25.64	368.1	...	3,640	10.6	6,297.8	6,370	...	17.10	17.20	0.535	Dry steam.	A
1883	128.8	28.00	368.1	...	3,640	Dry.	...	6,370	...	...	14.43	0.561	Dry steam.	B
1883	128.8	24.16	344.5	...	3,640	10.0	5,016.0	5,069	...	19.7	19.90	0.491	Dry steam.	C
1883	128.8	28.00	344.5	...	3,640	Dry.	...	5,069	...	...	14.77	0.548	Dry steam.	D
1883	7.03	25.60	393.0	...	3,600	85.0	8,086.0	...	...	26.35	...	0.606	Actual.	E
1883	5.46	23.75	392.0	...	3,000	89.0	7,076.0	...	...	36.68	...	0.589	Actual.	F
1883	8.45	24.75	295.5	...	3,000	93.7	6,063.0	...	...	29.48	29.48	0.649	Actual.	G
1883	1.75	24.35	307.5	...	3,300	76.5	8,340.0	...	...	27.10	...	0.639	Actual.	H
1874	100.4	25.30	...	106.4	2,500	12.0	6,220.0	...	58.5	38.4	...	0.591	Actual.	I
1874	100.0	28.8	...	98.8	2,468	0.981	3,440.0	3,310	34.85	22.85	20.00	0.443	Dry steam.	J
1874	100.0	0.0	...	98.8	2,525	12.5	6,065.0	6,142	61.8	40.58	41.10	0.414	Dry steam.	K
1750	135.5	25.25	416.9	...	3,600	26.9	6,400.0	6,575	...	15.35	15.78	0.597	Dry steam.	L
1750	135.5	28.00	490.9	...	3,600	Dry.	...	6,575	...	...	13.39	0.669	Dry steam.	M
1750	135.0	26.57	321.7	...	3,600	21.0	5,133.0	5,245	...	16.30	15.96	0.558	Dry steam.	N
1750	135.0	27.36	233.4	...	3,600	25.0	3,774.0	3,850	...	16.17	16.59	0.512	Dry steam.	O
1750	135.5	25.25	419.6	...	3,200	26.9	6,400.0	6,575	...	15.26	15.67	0.604	Dry steam.	P
1750	135.5	25.25	407.6	...	2,800	26.9	6,400.0	6,575	...	15.71	16.13	0.587	Dry steam.	Q
1750	135.5	25.25	384.9	...	2,400	26.9	6,400.0	6,575	...	16.63	17.08	0.554	Dry steam.	R
1750	135.5	0.25	67.0	...	900	110.0	6,874.0	...	...	102.6	...	0.175	Actual.	S
1750	0.25	26.01	167.5	...	1,300	110.0	6,874.0	...	...	41.04	...	0.438	Actual.	T
1750	0.25	26.01	237.6	...	3,500	110.0	6,874.0	...	...	28.94	...	0.620	Actual.	U
1750	0.25	26.01	227.8	...	4,000	110.0	6,874.0	...	...	30.19	...	0.595	Actual.	V
1750	124.7	26.49	343.0	...	3,600	27.4	5,165.0	5,305	...	15.06	15.47	0.578	Dry steam.	W
2044	125.0	26.49	295.0	...	3,600	Dry.	...	4,665	...	...	15.82	0.566	Dry steam.	X
2044	125.0	28.00	325.0	...	3,600	Dry.	...	4,665	...	...	14.36	0.568	Dry steam.	Y
2044	125.0	26.49	240.0	...	3,600	Dry.	...	3,990	...	...	16.62	0.539	Dry steam.	Z
2044	125.0	26.49	180.0	...	3,600	Dry.	...	3,260	...	...	18.10	0.495	Dry steam.	AA

bine proper, but is bolted to an entrainer or separator which in itself is bolted rigidly to the bedplate (see Fig. 9). From this entrainer the steam is led vertically through a flexible steel pipe which leads to the top of the turbine casing. In this way no outside stresses due to the heavy low-pressure steam-supply piping are thrown on the turbine itself. Furthermore, by the introduction of this additional entrainer, drier steam is obtainable in the turbine than would be the case were the inlet piping connected directly to the turbine proper.

Throughout the design of the machine special attention was paid to obtaining the best possible water rate consistent with a compact and reliable mechanical design, and the figures given in the table of some recent tests show some remarkably high efficiencies for turbines of such small power.

The last degree of efficiency can be obtained only by a careful study of the correct areas through the blade passages and by special attention to the finish of the blading in the wheels to eliminate friction and eddies. By the adoption of drawn material for the blading, true areas can be obtained. This machine, as built, conforms closely to calculations, as has been evidenced repeatedly by the careful observation of pressure drops through the various stages, these falling almost exactly in line with the calculated drops. The adoption of polished wheels further enhances the efficiency.

Some interesting experiments were carried out to determine the effect of rough and finished blades and wheels. Fig. 10 shows two curves, one with rough drop-forged blades, the other with blades polished to a true section and with the wheel polished.—"Power."

GEARED MARINE TURBINES.

At a recent meeting of the Birkenhead and Liverpool Foremen's Mutual Benefit Society, Mr. J. Hamilton Gibson, the engineering manager of Messrs. Cammell, Laird and Co., gave some interesting data as to the type and dimensions of the teeth used in recent important installations of geared turbines. It might be thought, he said, that very heavy teeth would be required in cases in which 15,000 horsepower were being transmitted per shaft. That, however, was not the case, as for all large powers the standard pitch was only 0.583 inch, so that the teeth were actually less than 5-16 inch thick. That fine pitch led to smooth running, but the wheel and pinion must be held accurately in position and the pinion thoroughly supported. The teeth were all involutes, with a short addendum and large radii at the roots. By adopting the involute form all gears could be cut with the same standard hob, and, further, the oil clearance could be adjusted by opening or closing the distance between the wheel centers by a few thousandths of an inch. With involute teeth such an adjustment did not in any way effect the proper meshing of the teeth. In ships propelled by geared turbines the propeller thrust had to be taken by a thrust-block, and Mr. Gibson said that when it was proposed to take on single collars the screw thrusts of the *Buenos Aires* and *Monte Video*, which developed 5,000 horsepower, many engineers predicted disaster; but the Michel system had proved perfectly successful, the engineers reporting that during the whole run to Buenos Aires the "thrust-bearings ran quite cool, with the chill not off."—"Engineering."

MARINE ELECTRICAL ENGINEERING.*

THE REMARKABLE DEVELOPMENT IN THIS BRANCH OF ENGINEERING.—INSTALLATION ON THE UNITED STATES COLLIER "JUPITER."

By H. A. Horner.

Although it is to be expected that electrical marine applications should be rapid, the developments in this branch of engineering in the last five years are remarkable. Minor applications which were in the experimental stage three years ago are now regular requirements for the more severe and larger service. Such equipments as electric bilge pumps, electric steering gear, etc., though they were utilized on certain foreign ships-of-the-line years ago, held no place in our practice until recently. It is a long span from Jacobi's electrically-propelled boat of 1838 to the United States fleet collier *Jupiter* of 1913; but the gap of years contains progress in the art that will stabilize the application.

GENERAL INSTALLATION.

This survey of advancement must be brief. Following the usage of the time, 80 volts direct current was first adopted. The battleships *Kearsarge* and *Kentucky* building in 1899 were equipped with a 3-wire system using 80 and 160 volts. This precedent led to the standard voltage of 125 volts adopted in 1902. Last year the voltage was raised to 230 volts direct current.

This increase of potential, caused by the expansion of power applications, is disadvantageous to the lighting system and searchlights. The development of the 230-volt tungsten lamp in this country is not such as to insure its adoption at the present time, and searchlight lamps require an arc voltage of 50 to 55. These conditions have brought about the consideration of a system employing either 3-wire generators or 2-wire generators and balancers. Thus there is provided 230 volts for the power system and 120 volts for lighting and searchlights. A standard 3-wire generator has not been adopted and the proposition, therefore, must be classified experimental.

Early in 1913 the standard-conduit installation which had for some years been undergoing modifications and experimentations gave place to lead-covered steel armored conductors. This required new designs and methods for its proper installation, entailing changes in all the appliances, fittings, and fixtures, throughout the equipment. It has led to a great reduction in grounds, it reduces the space and weight required for installation, it allows for further expansion, and incidentally presents a less obtrusive appearance.

Much improvement has been sought in the use of special insulating compositions to supersede porcelain. Various types of asbestos lumber, ebony asbestos wood, and like patented preparations are now generally required. Control panels, interior fittings, etc., are constructed of such materials and at the present time seem to give satisfaction. These experiments have led to the requirement of some such material for the main switchboards superseding slate for this purpose. As this is a recent requirement no switchboards of large size have been manufactured and the questions involved rest in the hands of the designer.

As the tungsten lamp has improved in land practice it now becomes acceptable in marine work. As stated above the voltage (120) is still maintained the same because of the tungsten lamp. Vibrations in certain

*Presented before Section D of the American Association for the Advancement of Science at the Philadelphia meeting, December 30–31, 1914. Mr. Horner is chairman, sections committee, American Institute of Electrical Engineers, Philadelphia, Pa.

parts of the vessel, naturally in the vicinity of the engine spaces, is severe, and carbon lamps cannot be entirely eliminated. The introduction of the tungsten lamps has awakened a renewed interest in the principles of good illumination, and experiments are now under way with prismatic globes, special reflectors, etc.

Many of the above points, and perhaps all combined, have brought improved methods of light distribution and the manner of controlling light. This in turn has caused more economical methods of running feeders to centers of distribution and so produced an installation that is far superior to the old methods in cost of upkeep and reliability of service. Incidentally it has tended to simplicity which materially affects the cost both first and last and points to a hoped for standard.

The old automatic magnet control of the arc has now been replaced by the motor-operated lamp. Experiments with different compositions of carbon so as to produce greater intrinsic brightness at the source is in progress at the present time. The foreign manufacturer is employing an impregnated carbon which permits a higher potential at the arc, 70 to 75 volts instead of 50 to 55 volts. A lamp using the alcohol spray is under consideration. This lamp is of interest due to the increased illumination. A small positive carbon is used and high temperature of the arc is effected by the use of the alcohol spray. It is understood that impregnated carbons are also employed. It is not known whether any attempts have been made to utilize high-power gas-filled tungsten lamps for this purpose, but it is supposed that such a field would be promising. The pendulum swings between the advantages of remote electrically-controlled searchlights and hand or mechanically-controlled lights; but at this time the electrically-controlled light is not in favor.

POWER SYSTEM.

The steady improvement and continued use of contractors for the contol of motors has increased the use of automatic control for a varied number of applications. Dynamic braking, which quickly brings the motor armature to rest, is asserting its certain importance in many equipments.

Electricity is now solely depended upon for weighing the anchor and, although the results of this application for all conditions of service are not known, it is reasonable to believe that the application will warrant its continuance.

Experimental work for the last five or six years has brought electrical steering gear into what appears to be a permanent requirement. The older systems tried did not give satisfactory results and it was not until the development of the contactor that experimental work was again resumed. There are now two systems designed which will be shortly tried in service. One is based on the use of two motors controlled by contactors; the other based on one motor controlled by a motor generator on the Ward-Leonard system. Any type of electrical system is today paralleled by the old steam system, but the indications are that the electric gear will supplant the steam.

The introduction of oil fued instead of coal has changed the electric forced-draft fan to a steam turbine-driven fan, due in part to the fact that high pressure and low volume of air is required and also to the fact that the exhaust from these fans aids the heating of the boiler-feed water.

The main drainage pumps are electrically driven, but the application to other main engine auxiliaries has not increased generally. There is a movement toward rotary steam pumps instead of the old reciprocating pumps, which doubtless will lead in the end to the electric motor.

INTERIOR COMMUNICATION.

The old types of incandescent-lamp instruments have almost entirely disappeared and their place taken by the direct-current motor-operated instrument. The communicating systems as a whole have increased

greatly in extent. No absolute measure of this expansion can be given, but it is about 60 per cent.

One of the most important and interesting developments is that of the gyroscopic compass. Since 1852, when Foucault in his laboratory studied the physical action of the gyro and laid down its laws, scientists and inventors endeavored to produce for it some commercial usefulness. It is not the purpose of this paper to describe or narrate the history of this application. Those who are interested will do well to read H. C. Ford's paper entitled "The Electrically-Driven Gyroscope in Marine Work," read before the American Institute of Electrical Engineers at Detroit, June 23, 1914. It answers the intention of these notes of record that recent improvements in the essential point of quickly damping the oscillations has advanced the practical adaptability of this instrument. A demonstration of this improvement was given by Elmer A. Sperry before the 300th meeting of the American Institute of Electrical Engineers, held in Philadelphia, last October.

The improvements in wireless telegraph apparatus have been rapid and important. Most of these advances have been recorded elsewhere and need no further comment. In general the lines of improvement tend to the use of a continuous wave in preference to the undamped wave. Much success has been achieved by the quench-spark system. Matters of practical installation are more carefully looked after, and the value of wireless as a safety factor has not been exaggerated by the press.

Perhaps the most important advance in the protection of vessels occurred in the early part of this year. For many years and by many minds the transmission of sound signals through water was given special consideration. Water as a medium of signalling has many more desirable characteristics than air. The submarine bell attached to the buoy, the lightship or the shore could emit signals such that a vessel equipped with water-tank and receiver telephones could evade those places in which there was danger and so be protected in foggy or thick weather. But there remained no under-water means by which a vessel could itself communicate either with the shore or with another vessel. The submarine telegraph oscillator is the work of Prof. R. A. Fessenden, and provides for under-water inter-communication by the telegraph or telephone. So much interest surrounds this apparatus that it seems desirable to quote from an abstract of R. F. Blake's paper read before the American Institute of Electrical Engineers at Philadelphia, Oct. 12, 1914. "The apparatus consists of an oscillating electric-motor generator which has a strong electro-magnet surrounding a central core on which is an alternating-current winding. This copper tube is attached to a large diaphragm. When the alternating current passes through the core winding it induces a current in the copper tube, which, being free to move, vibrates back and forth, thus setting the diaphragm in vibration. This oscillator can also be used as a receiver." From this brief description it can be seen that a most important field is now covered by this apparatus.

ELECTRIC DRIVE.

Since Davenport, the Vermont blacksmith, exhibited his electric motor in London many attempts have been made to apply electricity to the propulsion of vessels. Many will recall the electric launches at the Chicago World's Fair in 1893. Abroad some light-draught river craft were built and so equipped. A few years ago two electrically-driven fire boats for the city of Chicago were successfully operated. A vessel built in England to ply in the Welland canal was equipped with electric motors and Diesel oil-engine generators. However the first practical application to a seagoing vessel is the United States fleet collier *Jupiter*.

The keel of the *Jupiter* was laid at the navy yard, Mare Island, Cal., on Oct. 18, 1911. The *Jupiter* was placed in commission April 7, 1913. The vessel was designed for a speed of 14 knots, developing 5,500 S.H.P.

with a load displacement of 19,300 tons at a draught of 27 feet 8½ inches. Her performance showed a speed of 14.99 knots developing 7,151.9 S.H.P. with a displacement of 19,452 tons at a draught of 27 feet 7½ inches. She has been in service over a year and a half, during which time she has had two trial trips, performed the regular functions of a collier, steamed 14,000 miles, and successfully arrived at the Philadelphia navy yard on a continuous trip from San Francisco, stopping only in the Panama canal in order to view the operation of this great engineering project. During this service only two repairs have been made to the propelling apparatus. One of these could have happened to any type of engine; the other was of such minor importance that it need not be mentioned.

SUCCESS OF "JUPITER."

The success of the *Jupiter* is many fold. The fuel economy is shown to be 25 per cent. better than the best of light vessels equipped with other methods of propulsion. The propellers are exceedingly efficient, giving the same speed as sister ships with a reduction of 300 to 800 horsepower. The maneuvering qualities of the vessel are markedly superior due to the rapidity of reversal and also to the fact that full power is available for backing. The ship is about 542 feet long and about 65 feet in beam and would be extremely unwieldly if it were not possible to aid the rudder by means of the propelling motors. There are many more advantages, but the all-important matter is well summed up in the conclusion of a paper by Lieut. S. M. Robinson, U. S. N., read before the Society of Naval Architects and Marine Engineers, Dec. 10, 1914. "After all, the greatest test of the satisfactory working of any machinery is whether or not the men who are actually handling and caring for it are pleased with it. If this test applies to the *Jupiter's* machinery it certainly is an unqualified success. In particular is this true if the matter is referred to the coal passers in the fireroom who have to handle much less coal than do the men on sister ships. The ship can make her contract speed of 14 knots without using forced draft at all."

The success of the *Jupiter* has led the United States Government to extend this application, and electric propulsion is now authorized for the battleship *California*, now building in the New York navy yard. This equipment naturally will be an advancement over that of the *Jupiter*. With this development many more advantages may be expected, as electricity provides ready means for the accurate measurement and determination of propulsion factors. As the application grows many problems of the naval architect and marine engineer will be reopened, doubtless to the betterment of water transportation. There remains to the shipbuilding art today many questions that are settled upon theory because facilities are not given to practically record the conditions of performance. This obstruction will be greatly removed by the application of electricity.

PRINCIPLE OF ELECTRIC PROPULSION.

The first announcement of the principles of electric propulsion of naval vessels may be found in a paper read by W. L. R. Emmet before the Society of Naval Architects and Marine Engineers, Nov. 18, 1909. It is to Mr. Emmet that credit should be given for the introduction of what will doubtless prove to be the best and safest means of ship propulsion; to say nothing of the increased possibilities for the advancement of the art of ship design.

The last half decade is replete with progress in the marine applications of electricity. It is noteworthy that these applications have followed land development in the electric field. From an auxiliary of little importance except lighting, electrical applications of power have grown until now we find electricity entering into the main design of the vessel. The marine

engineer is willing to admit that an electric motor will drive a ship, but imagine his astonishment when he finds in the future to what extent the full effect of this application will lead.—"The Marine Review."

ELECTRIC SHIP PROPULSION.

Before a joint meeting of the Western Society of Engineers and the American Institute of Electric Engineers, W. L. R. Emmet, of the General Electric Co., gave an interesting talk on the above subject on the evening of April 26. The development of the high-speed turbine paved the way for electric ship propulsion. Its application in this field had been long foreseen. Mr. Curtis had worked for two or three years on the problem, and since 1900 Mr. Emmet had spent much of his time on the turbine. About six years ago he had first approached the Navy with a view to equipping battleships for electric drive, but at about the same time the question of reduction gearing had been brought to the front and the Navy had been impressed to the extent that the collier *Neptune* was equipped with turbines and reducing gears. The excellent results obtained aroused interest in the general question of reducing the speed between the turbine and the propeller, and as a result Mr. Emmet secured the contract to equip the *Jupiter* electrically. During the two years this ship has been in service it has made a wonderful record. Results 20 per cent. better than from any ship afloat have been obtained, and the equipment is as good as new. The turbines run regularly on a water rate of 11 pounds per shaft horsepower hour, which may be compared to 14 pounds, the best obtainable from a triple-expansion-engine-driven vessel. Naturally, electric propulsion gained in favor, and about a year ago it began to be thought of seriously for battleships. As the advantages of the electric drive increase with the power required, Mr. Emmet had been particularly anxious to equip a battleship, and only within the last few days the contract for the *California* had been closed. An estimate on the cost of installing electric drive showed that a saving of $160,000 would be effected over the cost of the turbine equiment that had been previously planned. In these large powers all sorts of complications arise when the turbines drive the propellers directly or through reduction gearing. With the latter the power must be divided up between a large number of units, as there is a limit to the size and capacity of individual gears beyond which it would not be safe to pass. In ships where the turbines drive the propellers directly there must be a compromise in speed. The turns made by the propellers are much too high, and the turbine runs at about a tenth of the speed it ought to have to give the best results. Besides, there is great complication of piping for high- and low-pressure turbines, and as the pressure in some of this piping is below the atmosphere, air leaks are liable to develop and lower the efficiency by reducing the vacuum. On the other hand, with the modern electric drive the loss cannot exceed 8 per cent. The apparatus is designed so that practically a constant water rate is maintained for all loads.

In the *Lusitania*, with a speed of 180 r.p.m. the propeller efficiency is 62 per cent. The turbines for the *California* will have a speed of 2,200 r.p.m. and deliver to the generator 75 per cent. of the available energy in the steam. In the latter the turbines will be simple, compact machines, while those of the *Lusitania* are enormous. By dropping the propeller speed of the *Pennsylvania* from 222 to 160 r.p.m., the efficiency would be increased 8 per cent., which would just counterbalance the loss by electric propulsion. Comparing the present equipment with an electrically propelled *Pennsylvania*, the efficiencies would bear a ratio of about 63 to 73 per cent.

Investigating the possibilities of reduction gearing held back electric propulsion. Parsons had condemned the latter and favored gears. Reduction gearing has proven successful on small ships running at moderate speeds. As the speed of the vessel increases, however, the ratio of reduction between turbine and propeller speeds becomes greater and the gear problem is more difficult. The General Electric Co. had become interested in gearing and developed a system which was installed on three freighters equipped with turbines. These gears may be applied to cases where electric propulsion is barred, but in the favorable cases the speaker could not imagine any arrangement of gears which would be anywhere near as good as electric drive.

Electric propulsion is to have a wide field of application. The company had recently figured on two large Russian cruisers and on a number for our own Navy. Mr. Emmet stated that he could reëquip the *Lusitania* and save $150,000 per year in the cost of coal. Electric drive for liners so far exceeds engines that the equipment would pay for itself in one or two years.

Slides were thrown on the screen showing the 20,000-ton collier *Jupiter* and its power-plant equipment. At 15 knots the vessel requires 7,000 horsepower. The generator is of simple and rugged construction and is not restricted as to voltage or frequency. It has a capacity very little greater than required by the motors, so that even a short circuit would not result in much injury. The motors are of the three-phase induction type, the stator having bar windings and the rotor a definite wound design provided with external resistance to be used when reversing. The governor is designed much like a tachometer with a system of fulcrums which can be moved in and out and varied through a wide range of speed.

For the *California* each turbine will have a maximum capacity of 18,000 shaft horsepower and on maximum load will require 170,000 pounds of steam per hour. The vessel has a displacement of 30,000 tons and a maximum speed of 22 knots, and yet each of the two turbines driving it is only 14 feet long. The motors are 12 feet in diameter by 11 feet wide. Consequently, the entire equipment occupies comparatively little space, and the first impression would be that it was designed for a tugboat or at least a vessel much smaller than the *California*. Even the auxiliaries will be electric driven, and the only steam piping entering the engine room will be the two leads for the main turbines.

The two turbines will develop a maximum of 36,000 horsepower, which is required to force the vessel to 22 knots. At 14 knots 7,000 horsepower is required. Performance charts showed that the water rates will remain practically constant over a wide range of speed. At 14 and 21 knots it was 10¾ pounds per shaft horsepower-hour, and for the range in speed between these two points it remained between 10 and 11 pounds. At a speed of 15 knots, 28½ inches of vacuum, no superheat and 190-pound gage pressure, the *Jupiter* showed a perfomance of 11 pounds per shaft horsepower-hour. These figures are exceptional and can be obtained only when both the turbine and the propeller are running at their most efficient speeds. By diminishing the excitation with the speed the efficiency is maintained and at the same time the torque is not reduced beyond that which is required. It is simply a case of diminishing the excitation until the propellers are turned at the right speed with the minimum amount of steam. One of the big problems is reversing, but it has been met by using high excitation while the change in direction is taking place.

With the reduction gear the great problem has been to equally distribute the load over the entire face of the gear. With a rigid gear most of the load is applied near the ends of the teeth. In the General Electric design this difficulty has been overcome by a gear made up of separate discs which will give sidewise and distribute the load over the surface.

A number of charts comparing the relative economy of engine-driven vessels, geared turbines and electric propulsion showed the following water rates per shaft horsepower-hour. For the *Vespasian*, with triple-

expansion engines, the water rate was 19 pounds; with geared turbines, 16 pounds; and with electric drive, 12.7 pounds. The *Cairngowan*, with triple-expansion engines, developed a shaft horsepower-hour on 17.3 pounds of steam, and the *Cairnross*, a sister ship with geared turbines, on 14 pounds. It was estimated that either vessel equipped with electric drive would develop a shaft horsepower-hour on 11.77 pounds of steam. The above figures tend to prove the assertion made by Mr. Emmet that electric drive over triple-expansion engines will reduce the water rate about one-third.

In the discussion it was brought out that as induction motors cannot run above synchronous speed, the propellers cannot race. Even in a heavy sea, with the propellers entirely out of water there is no vibration or any indication of a change in speed. As to the proper fields for reduction gearing and electric drive, Mr. Emmet made the general statement that in all ships requiring above 15,000 to 20,000 horsepower, gearing would make a poor comparison. In vessels requiring 10,000 horsepower and less and running at a low speed, reduction gearing would perhaps make the best showing. The field for electric drive is in large merchant ships and all battleships with the exception of torpedo boats and destroyers, where restrictions in weight prohibit its use.—"Power."

TESTING LUBRICATING OILS.*

SYNOPSIS.—An enumeration of the various tests to determine the qualities of an oil. Results obtained are only approximate.

A brief description of a desirable lubricant is that its viscosity should be the least possible which will allow it to stay in place and do the work. Summarizing the commonly desirable characteristics, they are:

1. The oil should possess cohesion.
2. It should possess the maximum possible adhesion.
3. It should be as far as possible unchangeable.
4. It should be commercially free from acid.
5. It should be pure, that is, it should be what it purports to be.

TYPES OF VISCOSIMETER.

The first to be discussed is the viscosity test, which is used to measure the internal friction of the oil, or, as an engineer might put it, the shearing modulus. This test is of value because a lubricant is really used to keep a shaft or journal and its bearing apart. The journal really revolves on a sheet of lubricant, an action which has been described as revolving on the molecules of the lubricant. The ease with which the molecules slide over one another therefore determines, to a certain extent, the friction loss in a bearing.

A fine example of the effect of the viscosity of lubricating oil is furnished by an experience in a certain spinning mill. The mill was operated with power derived from an engine carrying about the maximum load of which it was capable. The lubricant used on the spindles was changed to one which was supposed to be better. It was then found that the engine did not have power enough to drive the machinery in the mill; as a matter of fact, it was unable to start the machine in motion. Examination showed that the only essential difference between the two lubricants was the possession of higher viscosity by the new oil.

The measurement of viscosity of lubricating oils is in a certain sense unsatisfactory, because the results obtained with the different instruments which are available for making this test do not agree among themselves.

*Abstract of paper read by Prof. A. H. Gill before the Detroit Engineering Society, Mar. 19, 1915.

It is therefore customary to state the instrument which was used in determining any quoted viscosity.

One of the most commonly used viscosimeters is the Saybolt instrument. This is of the pipette type, having a tall pipette of rather small diameter immersed in a jacket which may be used for maintaining any desired temperature during the test. The test is made by filling the pipette to a certain point and noting the time of efflux, in seconds, which is taken as the measure of the viscosity of the oil tested. Or the so-called specific viscosity may be determined by dividing the time required for the efflux of the oil by the time required for the efflux of the corresponding volume of water. The Saybolt instrument was developed by the Standard Oil Co. and was used as a standard for many years, and is today.

The instrument most commonly used in Germany, and now coming into rapid use in this country by both the Government and individuals, is known as the Engler viscosimeter. This differs from the Saybolt principally in using a shorter pipette of larger diameter. It is used in the same way, but the specific viscosities as determined by the two instruments do not agree.

None of the commercial viscosimeters really measure the viscosity, because it can be shown that the tube through which a jet is discharged must have a length of from 175 to 200 times the diameter to give a true measure of viscosity. Any of the commercial instruments can, however, be standardized by measuring the times of efflux of standard solutions of cane sugar of glycerin. By such means the readings of these instruments can be interpreted in terms of absolute viscosity in dynes.

Numerous viscosimeters made of glass have been tried, but unfortunately no two glass instruments can be made exactly alike except at prohibitive expense. For this reason, the glass pipette once used as a standard by the Pennsylvania R. R. was abandoned. It should, however, be noted that a glass pipette, calibrated with glycerin as above described, can be used.

The viscosimeters just mentioned are all of the efflux variety, but there are numerous other forms available, and some of them are particularly well adapted for testing the viscosity of certain commercial products other than oils. For instance, the retarding effect exerted on a paddle revolved in a viscous liquid may be used as a measure of the viscosity and is so used with varnishes, glue and paste. Another form consists of a cylinder suspended from a torsion wire. The retarding effect upon this cylinder when swinging back to normal position after a displacement can be used as a measure of viscosity.

It should be particularly noted that the viscosity varies rapidly with the temperature. It is therefore necessary to state the temperature at which any determination was made.

FRICTION TEST.

There is really no satisfactory test of the adhesive quality of a lubricant. It is commonly supposed to be determined by a friction test. This is made by measuring the frictional resistance offered to the rotation of a standard journal in a standard bearing when lubricated with the oil in question. The results obtained depend partly upon the viscosity of the lubricant and partly upon its adhesion. Modern research shows that viscosity tests show about as much as do friction tests, but this is not necessarily so, and must not be assumed to be universally applicable.

GUMMING TEST.

A third test, and one which is of great importance, is known as the gumming test. This is particularly applicable to petroleum oils and is used to indicate the extent to which the oil has been refined. It serves

indirectly to indicate the extent to which the oil may be expected to change due to oxidation when in use. Numerous opportunities have been offered to check the results obtained with this test and results obtained in practice with the same oils, and all of this experience tends to show the great value of the gumming test.

This test is made by putting a small quantity of the oil to be tested in a small glass vessel, such as a cordial glass, and then mixing with it an equal quantity of nitrosulphuric acid. A properly refined oil will show little, if any, change, but a poorly refined oil will be indicated by the separation of large quantities of material of dark color. This color is due to the oxidation of the tarry matter contained in the lubricant. Experience has shown that oils containing large percentages of tar absorb the most oxygen, that is, they are mildly drying oils.

The results obtained by the gumming test agree well with carbon-residue tests made by distilling to dryness in a glass or a fused quartz flask. The carbon-residue test has been found of great assistance in choosing a satisfactory cylinder lubricant for gas engines, as a large amount of carbon means trouble in the engine cylinder. The lowest carbon content mentioned by the author was 0.11 per cent. The oil giving this test showed no tarry matter when tested with nitrosulphuric acid. In general, a gas-engine oil should not contain more than 0.5 per cent. carbon as determined by the carbon-residue test.

FLASH, FIRE AND EVAPORATION TESTS.

When an oil has been found to have satisfactory viscosity and has given satisfactory results in a gumming test, it must next be checked for safety, that is, the flash and fire test must be made. The amount of volatile matter given off at the temperature at which the lubricant is to be used is often of great importance. A case is on record in which a serious mill fire was spread by vapors given off by the lubricant used in the various bearings. The oil used in this mill gave off 25 per cent. of volatile material when raised to 145 deg. F.

It is advisable to include an evaporation test with the flash test of lubricants. The evaporation test is made by exposing about 0.2 gram of oil at a proper temperature and determining the loss by weight in a given time.

The flash test is made by heating the oil slowly in a vessel surrounded by a proper bath and determining the lowest temperature at which a flame passed over the surface will ignite the vapors which are given off.

FREE ACID TEST.

It is generally conceded that lubricants should be practically free from acid, and the so-called free acid test is made to determine the extent of acid content. The mineral oils are agitated with sulphuric acid during the refining process for the purpose of removing tarry materials, and this acid must be practically all removed from the oil before it is put on the market. Oils may become contaminated with acid from another source as well; namely, the animal or vegetable oils which are occasionally mixed with them for the purpose of modifying their characteristics. A content of 0.3 per cent. of acid is generally considered the maximum allowable.

SPECIFIC GRAVITY.

It is often desirable to determine the character of the raw material from which a given lubricant was made. This can be done in the case of oils refined from petroleum by means of the specific-gravity test. Experience has shown that lubricants made from petroleum with an asphaltic base run from 7 to 10 deg. Baumé heavier than similar lubricants made from petroleum with a paraffin base.

In examining oils, it is well to bear in mind that the viscosity is easily increased by the use of a material known as oil pulp or oil thickener. This is really oleate of alumina, and while it brings up the viscosity, it does not give the greasiness expected when that particular viscosity was specified. At ordinary temperatures, a small quantity of this material will greatly raise the viscosity.

COLD TEST.

There is another test, known as the cold test, which is of value in some cases. If an oil is to lubricate a bearing, it must be fluid enough at the temperature of use to readily flow into that bearing. Many ruined bearings and some fires have resulted from the use of an oil which became too viscous to flow under the conditions of use. For such reasons, it is customary to chill samples of oil and to determine the temperatures at which they become too thick to flow readily.

IODINE TEST.

Tests other than those already described are often made on animal and vegetable oils. They are generally made for the purpose of determining whether the oil under test is what it is supposed to be. It is a simple matter to mix different animal and vegetable oils in such a way that they will give a product capable of passing any one or possibly two given tests, but it is impossible to make such a mixture successfully pass all of the tests which would be passed by the pure oil for which the mixture is to serve as a substitute.

The chemist is often at a great disadvantage in testing such mixtures, because there are no exact specific tests for some of the animal and vegetable oils. The presence of some can be determined absolutely, but unfortunately, this is not true of all.

The iodine test, by which is meant the determination of percentage of iodine absorbed by the oil under set conditions, has long been used to indicate the character of vegetable and animal oils present in a sample. At one time it was believed that the so-called iodine number was a constant for any one oil and that this test was therefore perfect. It is now known that this is not true, the iodine number varying with the condition of the material from which the oil was made.

It is a simple matter to determine the presence of petroleum oils in a mixture of oils with animal or vegetable origin. This is done by saponification, which serves to separate the petroleum oil, which does not saponify, from the others which do.

MAUMENE TEST.

There is a comparatively new test, known as the Maumené test, which gives results comparable with those obtained with the iodine test, but is much simpler and therefore more readily performed by the average individual. For this test, 50 grams of oil and 10 c. c. of sulphuric acid are placed in a beaker and slowly stirred with a thermometer. The maximum temperature rise which occurs is noted and used as an indication of the character of the oil.

TESTS ONLY APPROXIMATE.

It should be appreciated by the practical man that the tests of lubricating oils give only approximate results. Thus any one viscosimeter as ordinarily made will give consistent results on the same material at the same temperature, but different instruments of the same type and apparently exactly alike will give results on the same material which vary several per cent. Similarly, large errors are often obtained when using friction machines. With tests otherwise properly conducted, it appears that the

absorption of oil by the metal of the journal and bearing may be sufficient to cause appreciable errors. Tests have shown that it may take several hours to eliminate the effects of the last oil tested so as to get correct results with a given sample.

No rigid directions can be given for the choice of oils for given purposes. It is best to try various lubricants which can be purchased for any given lubricating problem until one is found which gives satisfactory results. This should then be completely tested and the results of the test should be used in writing specifications on the basis of which bids are to be asked. When the problem is handled in this way, the different prices asked for lubricants which will meet the same specifications will often be found most remarkable.—"Power."

"HAND FIRING SOFT COAL UNDER POWER-PLANT BOILERS."

Is the title of a paper just issued by the United States Bureau of Mines, as an aid to the firemen employed in manufacturing establishments throughout the United States.

The paper, which contains descriptions of methods of firing soft coal under power-plant boilers and of methods of handling fire so as to have the least smoke and to get the most heat from the fuel, seeks to meet the needs of the men, many without a technical education, who are employed in small plants of 1,000 to 2,000 horsepower capacity. For this reason the language used is plain and simple, and technical terms have been avoided as far as possible.

The publication under "General Directions on Firing Soft Coal" makes the following statements:

"When burning bituminous coal under power-plant boilers the best results are obtained if the fires are kept level and rather thin. The best thickness of the fires is four to ten inches, depending on the character of the coal and the strength of draft. The coal should be fired in small quantities and at short intervals. The fuel bed should be kept level and in good condition by spreading the fresh coal only over the thin places where the coal tends to burn away and leave the grate bare.

"Leveling or disturbing the fuel bed in any way should be avoided as much as possible; it means more work for the fireman and is apt to cause the formation of troublesome clinker. Furthermore, while the fireman is leveling the fires a large excess of air enters the furnace, and this excess of air impairs good efficiency.

"The ash-pit door should be kept open. A large accumulation of refuse in the ash pit should be avoided, as it may cause an uneven distribution of air under the grate. Whenever a coal shows a tendency to clinker, water should be kept in the ash pit. All regulation of draft should be done with the damper and not with the ash-pit doors.

"In firing, the fireman should place the coal on the thin spots of the fuel bed. Thin and thick spots will occur even with the most careful firing, because the coal never burns at a uniform rate over the entire grate area. In places where the air flows freely through the fuel bed the coal burns faster than in places where the flow of air is less. The cause of this variation in the flow of air through the different parts of the fuel bed may be differences in the size of the coal, accumulations of clinker, or the fusing of the coal to a hard crust. Where the coal burns rapidly, the thin places form.

"Before throwing the fresh coal into the furnace the fireman should take a quick look at the fuel bed and note the thin spots. In a well-kept fire these spots can be usually recognized by the bright hot flame. The thick places have either a sluggish, smoky flame or none at all. In order

to place the coal over the thin places the fireman should take a rather small quantity of coal on his scoop, for it is much easier to place the coal where it is needed with small shovelfuls than with large ones.

"The coal should be placed on the thin places in rather thin layers. If the fireman attempts to fill up the deep hollows in the fuel bed at one firing, the freshly-fired coal may fuse into a hard crust, thus choking the flow of air, causing the fuel to burn slowly and starting new high places. If the high places in the fuel bed are missed on one or two firings the hard crust at the surface will gradually burn through or crack, thus allowing more air to flow through, and the place will get back to its normal condition. Of course, if the high place in the fuel bed is caused by clinker the flow of air will not be free until the clinker is removed with the fire tool. Whatever may be the cause of the high places in the fuel bed, the fireman should remember that they are places where the coal does not burn. There is no use in placing coal on such a place.

"If the firings are too far apart the coal in the thin spots may burn out entirely, allowing a large excess of air to enter the furnace in streams. If those streams of air are not properly mixed with the gases from the coal, only a small percentage of the air is used for combustion, and most of it passes out of the furnace, depriving the boiler of considerable heat. If, for instance, air enters the furnace at atmospheric temperature, say 75 degrees F., and leaves the boiler at about 575 degrees F., it carries away the heat that was absorbed in raising its temperature 500 degrees F. This heat is lost to the boiler. Another loss of heat occurs when holes form in the fuel bed, because pieces of unburned coal fall through the grate when the fireman attempts to cover the holes with fresh coal. Therefore, in order to avoid the formation of holes, firings should be made at short intervals, particularly if, for any reason, the fuel bed must be kept thin."

Copies of Technical Paper 80 may be obtained by addressing the Director of the Bureau of Mines, Washington, D. C.—"The American Marine Engineer."

GALVANIZED IRON AND STEEL.

By John Hamilton Paterson, D.Sc.

The methods which have been devised for the protection of iron from corrosion are legion; and although considerable ingenuity has been shown in some of them, very few have stood the test of time. One of the oldest of these methods, and also one of the most effective, is that of coating the surface of iron with zinc, commonly known as galvanizing. This process was patented by a man named Crawford in 1837.

The original idea in galvanizing iron was merely to coat the iron with a metal which was less susceptible to atmospheric influences; and although this holds good in the case of zinc, there is another and much more important property which makes it a valuable protective agent. This lies in the fact that, if a piece of zinc and a piece of iron are kept in very close contact with each other in an atmosphere or under conditions likely to cause corrosion, the zinc will itself corrode, but will protect the iron in its immediate vicinity and reduce the corrosion on it to a minimum. It is for this reason that boilers have zinc bars fastened inside their water spaces, and that the stern-frames and plates of vessels which have bronze propellers have zinc plates screwed on to them. The phenomena which are responsible for the protective action of zinc are well understood, but are somewhat outside the scope of this article. A piece of tin, on the other hand, when kept in close contact with iron under conditions likely to

cause corrosion, will itself remain perfectly bright, while the iron will disappear rapidly. Iron coated with zinc, therefore, will remain free from corrosion after the surface of the zinc becomes broken or cracked; whereas iron coated with tin, as in tin-plate, will rapidly rust away as soon as the continuity of the tin surface is destroyed.

The galvanizing of iron is effected in three different ways, known respectively as the Hot-galvanizing process, Sherardizing, and Electro-galvanizing. Hot galvanizing is carried out by dipping the articles into a bath of molten zinc, after they have been completely freed from grease, rust or mill scale by treatment with hydrochloric acid. Galvanized sheets are sometimes rolled after this treatment in order to make the zinc adhere more strongly; other than this there is no further treatment necessary. A small percentage of tin is sometimes added to the zinc bath in order to give the finished surface a spangled crystalline appearance. Articles with fine patterns or corrugations on the surface and pipes or bars with threads cut on them cannot be treated in this way, as the threads or corrugations become filled up with zinc. Large and heavy articles, such as ships' plates and corrugated iron sheets, are galvanized exclusively by this process; in fact in England 90 per cent. of the total galvanizing is done by the hot method.

Sherardizing, introduced by S. Cowper-Coles, but first made a commercial success in America within recent years, is a treatment whereby the zinc is volatilized on the hot, clean surface of the iron, where it forms an adherent and sound coating. It is accomplished by packing the articles, after a thorough cleansing, in zinc dust, and heating them in a suitable receptacle to a temperature of about 350 degrees C. The method is only suitable for small articles, but can be successfully applied to threaded articles, such as bolts and nuts, as the coating is thin and even and sharply follows the outline of the article. Sherardized articles have a dull surface, which, however, can be easily polished on a buff.

Electro-galvanizing, although an old process, is only just becoming possible on a commercial scale. It is accomplished by plating the article with zinc by means of an electric current in much the same way as articles of jewelry are plated with gold and silver. It produces a good adherent coat, and like sherardizing, can be used for nuts and bolts, as it does not fill the threads. It suffers from the defect, however, that the zinc coat occasionally begins to corrode without any apparent reason, and it is also an expensive process to apply. Tubes and pipes cannot be coated on the inside by the electrolytic process.

In spite of the fact that a broken surface of zinc is capable of protecting iron from rusting, it is very important that the zinc should be free from cracks and strongly adherent to the iron. The moment the iron surface is exposed the zinc begins to disappear rapidly, and it is not long before so much iron is laid bare that the zinc cannot protect it and then iron corrosion commences. The phenomena which cause the union of iron and zinc to one another have been closely studied and are somewhat as follow.

When carefully cleansed iron is brought into contact with melted zinc or zinc vapor, the two substances unite and form chemical compounds with one another. In the case of iron galvanized by the hot process, examination under the microscope has revealed the existence of four separate layers in the galvanized portion. These are (1) pure iron, (2) a very thin layer of unknown compositions termed the binding alloy, (3) a layer of an iron-zinc compound having the chemical formula $FeZn_5$, and (4) a layer of zinc permeated by crystals having the formula $FeZn_7$. When a galvanized sheet is sharply bent, parting takes place at the surface of the binding alloy; but in blistering and flaking when heated, parting takes place between the third and fourth layers.

With sherardized iron the coating varies with the conditions of the

process and the composition of the zinc powder, from a thin layer of $FeZn_8$, with a more or less distinct layer of binding alloy, to a thick coat of iron-zinc alloys, with a surface layer of pure zinc.

With electro-galvanized iron, coats of very pure zinc or of zinc-iron alloys are obtained, and in all cases a very thin layer of binding alloy.

From the above it will be seen that, whatever the process used, there is a very close union between the zinc and iron when the galvanizing is carried out carefully, and articles properly treated will stand rough handling without serious damage. The successful use of galvanized iron has led, however, to its adoption in a large number of instances where it is worse than useless. Although zinc withstands the action of ordinary atmospheric influences better than iron, it is attacked with the greatest rapidity by slightly acid or alkaline atmospheres. When open fires burning coke are used, the sulphur in the coke becomes converted into sulphuric and sulphurous acids, and if the roof of the building containing the fires be constructed of galvanized iron it will not last more than a few months unless a protective coat of paint is applied. Galvanized-iron buildings in the neighborhood of chemical works will suffer the same rapid deterioration unless protected by paint. It is much cheaper and more satisfactory in cases like these to erect plain iron buildings and protect them with paint only.

Zinc is attacked fairly rapidly by sea water, and although it acts perfectly as a protector to iron when immersed in sea water it is very quickly eaten away when both iron and zinc are exposed. The British Admiralty make a practice of galvanizing the frames and plates of light ships such as torpedo boats and destroyers. The surface of a ship thus constructed is, however, far from being completely covered with zinc. The rivets, of course, are of naked steel and the edges of a great many of the plates have been cut after the plate has been galvanized, thus exposing a raw edge of steel. It is obvious that if such a ship were left in the water without a protective coat of paint the zinc would soon disappear around the rivets and about any exposed edges of steel, and the shell would then be subject to the ordinary dangers of corrosion. It is clear, therefore, that galvanized ships are only ships with a very expensive protective coat applied to them, which must itself be as carefully protected by paint as is the hull of any ordinary merchant vessel. The only obvious advantage in building a ship of galvanized plates is that it allows of longer periods between the times necessary for repainting. If there is any danger of the too rapid corrosion of the very thin plates of which the hull of a destroyer is made, then surely the use of galvanized plates along the air and water line would be an ample precaution, especially at such a time as the present, when the congestion of galvanizing works seriously interferes with the rapid building of destroyers.

Another danger in the use of galvanized iron is illustrated in the constant failure of galvanized-iron steam and water pipes in merchant vessels. It is rarely the case that these pipes are galvanized after they have been bent to the required shape and threaded for screwing together. The pipes are bent hot, and during the heating process the zinc is all burnt off both inside and out for a considerable distance on each side of the bend. This damage is usually concealed by a thin coat of aluminum paint. The threading of the ends of the pipes exposes, of course, the raw iron, and a portion of this projects beyond the union, generally without any attempt at covering it. Even with these exposed places the pipes would be, with a little attention, quite free from danger of corrosion, but the average engineer regards a galvanized pipe as one which can be safely neglected; and it is a very uncommon sight, even with winch steam pipes laid along the deck, to see them with a proper coating of paint.

In the case of wire rope the galvanizing of the strands of the rope has been of the greatest possible service in prolonging the life of the rope.

In old galvanized-wire ropes corrosion usually started in the center of the rope, and was the more insidious as its effects could not be seen from outside. The coating of zinc on the wire, owing to the methods now employed, is usually very perfect and of considerable thickness. There is little fear, therefore, of corrosion taking place in the center of such a rope during its life.

In conclusion, it is to be noted that, while galvanizing is an efficient method of protecting steel and iron, it is open to several objections. One of these is the tendency to put too much faith in its protecting powers, and to leave galvanized structures to take care of themselves without adequate protection for the zinc. Another is that, in putting together buildings or other structures with galvanized iron, it is impossible not to have here and there exposed parts of uncovered iron. Such parts are frequently inadequately protected, and act as centers from which corrosion may start. Finally, it must be remembered that zinc, while resisting ordinary atmospheric influences, is very much more rapidly attacked by abnormal acid and alkaline atmospheres than iron.—"The Shipbuilder."

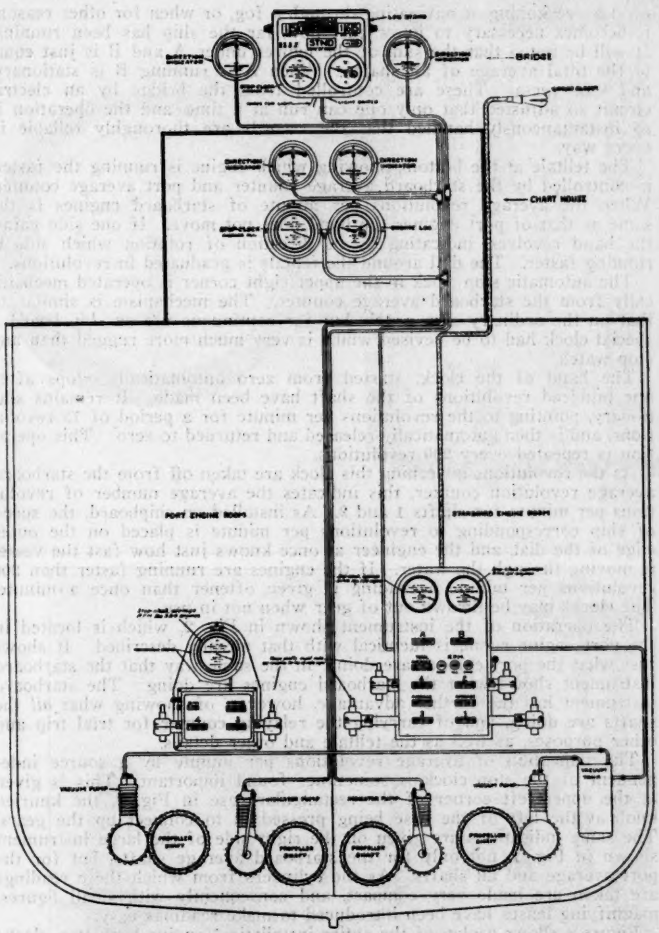
SPEED CONTROL ON DREADNAUGHT *PENNSYLVANIA*.

To promote harmony between the engine room and the bridge, and make it possible to get just that speed from the engines which is necessary for definite purposes, a system of instruments is to be used on the *Pennsylvania* which gives direct and immediate information at all times on the important subject of speeds of the several shafts, together with corresponding speed of ship and other similar matters. The essential parts of this installation, as fitted in the two engine rooms, are shown in Figs. 1 and 2, while Fig. 3 shows the interior of the instrument indicated in Fig. 1. The instruments on bridge and in chart house repeat automatically this same information. The apparatus was designed and built by the Cummings Ship Instrument Works, Boston, Mass.

While the instrument shown in Figs. 1 and 3 is complicated in appearance, it is really simply a combination of what in previous ships has been several independent instruments. The nine revolution counters have been in many cases fitted in three or more separate instruments; while the two stop clocks at the top have usually also been separate from the revolution-counter equipment, and the telltale is also frequently separate. The instrument as a whole, however, has many interesting features.

The four shafts of the *Pennsylvania* are operated from two engine rooms, two shafts in each. The counters from shafts 1 and 2 in the starboard engine room are run by the gears shown at the right of the instrument, and corresponding with the figures given under those shaft numbers. Similarly, shafts 3 and 4 in the port engine room are arranged on the left of the instrument. The revolutions of the shafts are transmitted to the instrument through the "one-way" gears at right and left respectively, these being arranged so that the counters always *add up*, regardless of the direction of rotation of the engine. The counters, therefore, always tell the total number of revolutions made, whether those were ahead or astern, and thus indicate the amount of work done by the engine.

By a combination of gears the indications of shafts 1 and 2 are averaged in the counter shown between those of the two shafts. It will be noted by a comparison with the figures that the number of revolutions indicated in that average is, in fact, a true average between the figure for No. 1 and that for No. 2. The use of positive-drive gearing for this purpose,



CUMMINGS ENGINE LOG SYSTEM INSTALLED ON U. S. S. "PENNSYLVANIA."

as shown in Fig. 3, avoids the errors inseparable from other methods of driving counters. In a similar way the average of the port shafts is given, and then by a combination of these two averages we obtain at once the average for all shafts, as shown just above No. 3 at the left of the instrument in Fig. 1.

The counters A and B are used for trial-trip purposes and for recording the distance run on any given course. This is important when running

on dead reckoning or navigating through a fog, or when for other reasons it becomes necessary to know just how far the ship has been running. It will be noted that the sum of the figures under A and B is just equal to the total average of all shafts. When A is running B is stationary, and vice versa. These are controlled from the bridge by an electric circuit so adjusted that only one can run at a time, and the operation is so instantaneously handled that the records are thoroughly reliable in every way.

The telltale at the bottom, showing which engine is running the faster, is controlled by the starboard average counter and port average counter. When the average revolutions per minute of starboard engines is the same as that of port engines the hand does not move. If one side gains, the hand revolves, indicating by its direction of rotation which side is running faster. The dial around the telltale is graduated in revolutions.

The automatic stop clock in the upper right corner is operated mechanically from the starboard average counter. The mechanism is similar to that on the ordinary stop watch, but for continuous use on ship board a special clock had to be devised which is very much more rugged than any stop watch.

The hand of the clock, started from zero automatically, stops after one hundred revolutions of the shaft have been made. It remains stationary, pointing to the revolutions per minute for a period of 75 revolutions, and is then automatically released and returned to zero. This operation is repeated every 200 revolutions.

As the revolutions governing this clock are taken off from the starboard average revolution counter, this indicates the average number of revolutions per minute for shafts 1 and 2. As installed on shipboard, the speed of ship corresponding to revolutions per minute is placed on the outer edge of the dial, and the engineer at once knows just how fast the vessel is moving through the water. If the engines are running faster than 200 revolutions per minute a reading is given oftener than once a minute. The clocks may be thrown out of gear when not in use.

The operation of the instrument shown in Fig. 2, which is located in the port engine room, is identical with that already described. It shows just what the port engines are doing, in the same way that the starboard instrument shows what the starboard engines are doing. The starboard instrument has the further advantage, however, of showing what *all* the shafts are doing, and of carrying the relaying counter for trial trip and other purposes, as well as the telltale and other details.

The indication of average revolutions per minute by a source independent of the stop clock is sometimes found important. This is given at the upper left corner of the rectangular case in Fig. 2, the knurled knob at the left of the case being pressed in to connect up the gears. The same indications are given on the right side of the large instrument shown in Fig. 1, not only for the starboard average shafts, but for the port average and all shafts. As the cylinders from which these readings are taken are made very compact, and consequently with small figures, magnifying lenses have been introduced to make readings easy.

Figure 4 shows a plan of the entire installation, engine logs, stop clocks and revolution-direction indicators being installed in chart house and on bridge, operated by vacuum and controlled by geared valve shown at middle of left side, Fig. 1. The engine logs show the distance traveled and are used in determining position of ship. They have been found to be the most reliable means for determining position, since they are based on revolutions of propellers. The system has been worked out in such detail that it is quite unnecessary for the navigating officer to telephone to the engine room, or communicate in any other way, to learn what the engines are doing. The information is all before him in the automatic

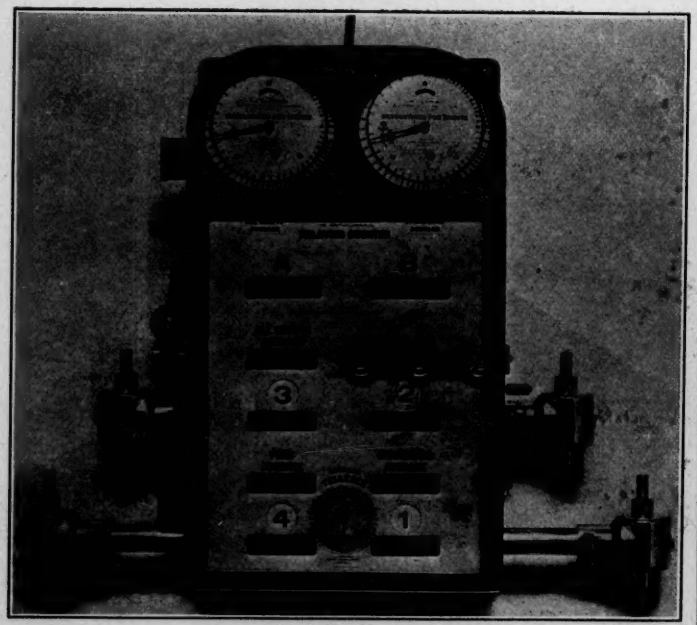


FIG. 1.

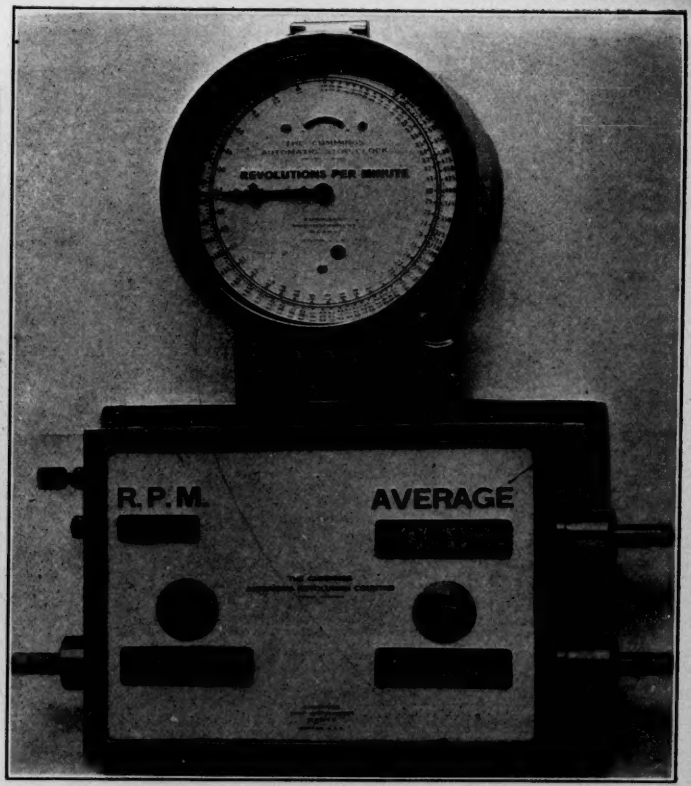


FIG. 2.

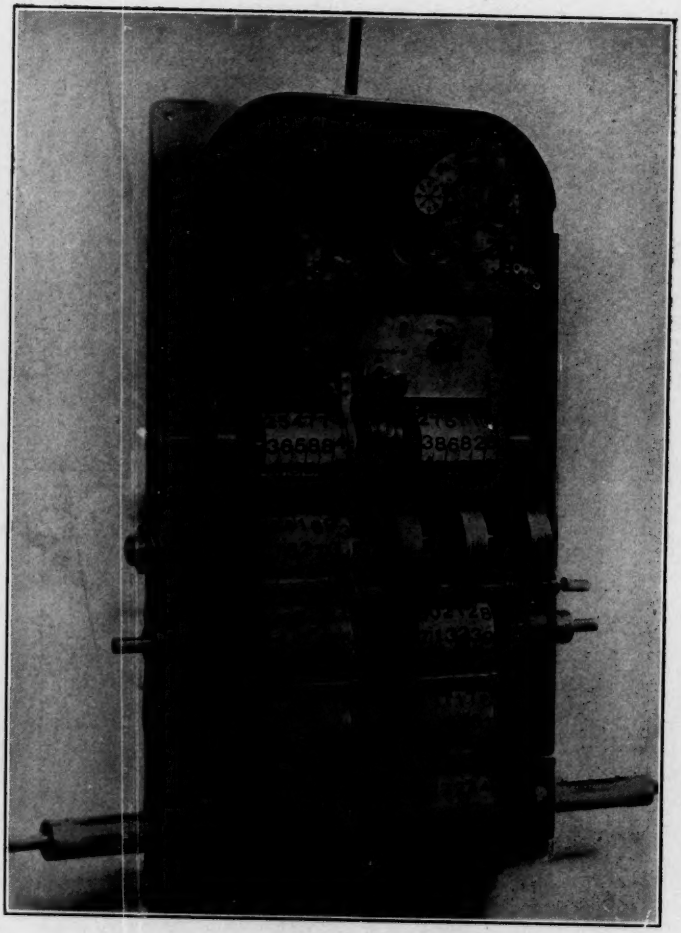


FIG. 3.

instruments—he can see at a glance just what the conditions are, and can take what steps may be necessary to produce the result he desires.

Equipment similar to the *Pennsylvania's* has been fitted on practically every dreadnaught in the United States Navy and on many of the earlier ships. It happens, however, that in the other cases the system has not been carried out to quite the same degree of thoroughness, although the major items of the information are transmitted automatically to the bridge in all cases.—"International Marine Engineering."

PAPERS AT THE INSTITUTION OF NAVAL ARCHITECTS.

A CONTRIBUTION TO THE THEORY OF PROPULSION AND THE SCREW PROPELLER.

By F. W. Lanchester.

The investigations of this paper are based on the idealized conception of a propeller, or "actuator," in which the fluid is assumed to be acted on by forces distributed over a defined surface normal to the axis of motion, representing in sum the magnitude of the total thrust, the said forces being parallel to and opposite in sense to the direction of motion.

Dr. Froude has stated that one-half the acceleration must take place before the propeller and the other half behind it. This view has been attacked by Prof. Henderson and been subsequently further discussed by Dr. Froude.

The present writer reviews the theory from its foundations in order to minimize uncertainty as to the ultimate interpretation of results. While disputing Dr. Froude's conception as to the possibility of transforming the efflux system of flow into a propulsion system, it is admitted that such is strictly analogous. Attention is called to the fact that the Froude actuator must be a plane surface acting normal to the axis of propulsion, and from the analogy of the flow from a re-entrant nozzle which the writer propounds it is suggested that Dr. Froude's results are misleading. The result obtained is that the velocity communicated before traversing the actuator is less than that communicated subsequently. Having established the view that the motion precedent to traversing the actuator contributes to the propulsion reaction, a consideration of the Froude actuator is entered upon independent of the efflux theory. For simplicity sake the actuator is defined by a circle. Roughly speaking, the forces of propulsion will produce a local contraction of the stream before passing through the actuator, followed by an increased contraction which continues until the ultimate contraction of the wake stream is reached, surface discontinuity with the outer flow existing throughout. Theoretical hydrodynamic reasons are given in refutation of the supposed re-expansion of wake stream abaft the actuator; any such phenomenon must be attributable to inter-diffusion or mingling between the wake stream and the surrounding fluid.

In introducing a modified hypothesis, it is assumed that the surface of discontinuity between the wake stream and the surrounding fluid degenerates rapidly into a number of vortex rings. Substituting a series of impulses for continuous action, each impulse will give rise to a number of vortex rings which will represent a definite quantity of momentum. A consideration of momentum thus engendered shows that the existence of counter wake does not affect the momentum quantity in analysis. It is claimed that this vortex emission theory directly applies to means of propulsion as at present known.

From consideration of artificial conditions which are regarded as approximating to actual conditions four deductions are reached, namely, that the momentum communicated to the fluid is quantitatively that which appears in the slip stream; the work done on the fluid by the actuator is quantitatively that which ultimately remains as kinetic energy in the slip stream; the kinetic energy in the counter wake system may be considered as a conservative system; both the kinetic energy and momentum imparted to the fluid forward of the actuator contribute to propulsion.

In accordance with the conditions assumed, more than half the acceleration is communicated to the slip stream in front of the propeller, and under the worst conditions the whole acceleration may be ragarded as antecedent.

The author concludes that, if by any mechanical device a continuous stream of vortices could be generated, the actuator will need to operate on fluid moving at a speed half way between the velocity of the general stream and that of the maximum imparted to the vortices; that is to say, Dr. Froude's result is arrived at.

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FURTHER MODEL EXPERIMENTS ON THE RESISTANCE OF MERCANTILE SHIP FORMS, AND THE INFLUENCE OF LENGTH AND PRISMATIC COEFFICIENT ON THE RESISTANCE OF SHIPS.

By J. L. Kent.

Part I of this paper is a continuation of that read before the Institution by Mr. G. S. Baker last year, and gives the results of a further set of model experiments made to determine the effect of shape of area curve on resistance.

Part II gives a general idea of the manner in which the resistance of a ship changes with alteration in form and length. The wave contours at different speeds have been taken for a series of models, and from them an attempt has been made to define some of the conditions which will promote heavy or light wave making.

The results of methodical model experiments in other tanks have been examined and some conclusions drawn from them on the effect of certain changes in form.

Part I.—As the result of experiments made on ten models of identical length, breadth and draught, but varying either in form or fullness, the following findings are arrived at:

Varying bow prismatic coefficient: At low speeds the finest entrance is slightly worse than either the medium or full forms; but for higher speeds it is better than either of them. For higher speeds, the fuller the entrance the worse is the result. At the highest speed to which the full entrance could be adopted, there is a loss of 20 per cent. in power over that for the finest entrance, with a gain of only 7 per cent. in displacement.

Varying stern prismatic coefficient: The results indicate that the fullness of stern has not such an influence on the resistance of ships as fullness in bow has, and although slightly greater resistance may accompany increase in after prismatic coefficient, the larger displacement obtained may outweigh the small disadvantage as regards resistance.

Varying shape of entrance: A hollow-line bow is found to be best for the particular model tried over the range of speeds possible for a ship of the given fullness, but the medium-line bow is never more than 3 per cent. worse. The straight-line bow is much worse for all speeds possible for any ordinary vessel of the said form.

Varying shape of run: The medium-line stern gives better results than either a hollow or a straight line aft.

Part II.—Dr. R. E. Froude has described the "principal wave-initiating operations" of the entrance and run of any ship which give rise to two systems of transverse waves, and he has shown that the coincidence of a bow-wave trough with a stern-wave trough will cause increased wave resistance, whilst a bow-wave crest coinciding with a stern-wave trough causes diminished wave resistance.

The purpose of the present communication is to give the results of experiments made to determine the effect of form of bow on the resistance of the model due to all possible phase difference between the bow and stern-wave systems. Two series of models, each consisting of an entrance and run joined together by different lengths of parallel body, eleven models in each series, have been tested. The difference between the two series was in the shape of the water lines of the entrance, one set having a straight waterline entrance and the other being hollow. In all other respects the models were alike. The bow wave for all models of the same set at the same speed was found to remain unaltered, irrespective of the length of parallel body. As the stern is brought nearer the bow by cutting out the parallel body, the growth of the primary trough of the stern-wave system, as a trough of the bow-wave system passes through it, becomes apparent, and its effect on the residuary resistance is most marked. The humps and hollows of the residuary resistance curves are caused by interference between the bow and stern-wave systems. The humps are evidence of an increased waste of energy, and the wave profiles show that this is due to an increase in the heights of the resultant system of waves formed by the model.

Part III.—To trace the effect on the resistance of a ship of a change in the prismatic coefficient or length, an examination of the methodical series of model experiments published by Messrs. R. E. Froude, D. W. Taylor and G. S. Baker are made. The effect of varying prismatic coefficient for ships of 450 feet length and 10,000 tons displacement from Froude's A series is that an increase in residuary resistance accompanies increase in prismatic. From a comparison of Froude's A and B series, the conclusion is arrived at that for ships having the same displacement, length, prismatic and midsection coefficients, the residuary resistance increases as the ratio $\frac{\text{Breadth}}{\text{Draught}}$ increases. From Taylor's experiments it is borne out that increasing the midship section coefficient, keeping the same prismatic coefficient, reduces residuary resistance for the same length ship.

LAW OF FATIGUE APPLIED TO CRANKSHAFT FAILURES.

By C. E. Stromeyer.

The author read a paper last year on the "Elasticity and Endurance of Steam Pipes," and in the present one he deals with crankshafts on the same lines. This problem is far more complicated than that associated with steam pipes, and the results arrived at are not precise. Thus, although crankshaft failures may be due to insufficient strength for ordinary working conditions, they may also be due to the shafts being out of line or to the bearings having become overheated and suddenly drenched with cold water.

The sudden cooling of a hot crank pin or journal is of course likely to lead to very serious results, for the cold water has access only to the fillet to which the violent contraction is confined, and as this is professedly the weakest part of a crankshaft, cracks are most likely to occur here. The fact that cracked crankshafts can safely be run for a few months suggests that these cracks are only surface deep and may therefore be produced by the sudden drenching of the hot shaft in cold water. The

other disturbing factor in this inquiry is the alignment of the shafting. The bending of shafts may be due to a settling down of the engine, to discontinuity in the ship's structure, to unequal loading of the ship or wearing of the brasses or to the slacking back of main bearing bolts. The bending moments in crankshafts are easily calculated if the following assumptions are made, as has been done in the paper. The mean indicated load on each piston was assumed to be constant throughout the stroke, the inertia of the moving masses counteracting the irregularities of the diaphragm. On dealing with crankshafts as though they were continuous —a most laborious process—it was found that the after crank was subjected to weaker bending stresses than the forward one and yet most of the failures occurred in the after crank. It was also found that very slight changes in the alignment increased the stresses, and as it seemed probable that with continuous working of an engine the bearings would wear down in such a manner as to equalize loads, this condition was presumed to be the prevailing one. In other words, both crankshafts were treated as if they were connected together by flexible couplings. Under these assumptions the bending moments of the several fillets were calculated and also the torsion moments.

The resisting moments were more difficult to deal with, for the corners of the crank had to be treated as being curved beams, and the author had in all the cases embraced in the paper recorded the radii of curvature of the fillets. Their influence on the resisting moment of cylindrical sections is proportional to the square root of the radius r of the fillet to the diameter d of the shaft, so that the strength of the curved beam of circular section may with reasonable accuracy be expressed by the formula

$$\mathrm{M} = \frac{\pi}{32} \times \mathrm{S} \times d^3 \sqrt{r},$$

where S is the maximum stress; and for an elliptical section the formula would be

$$\mathrm{M} = \frac{\pi}{32} \times \mathrm{S} \times b \times a^2 \times \sqrt{r},$$

where a and b are the major and minor axes of the ellipse.

The above sketched method of estimating the stresses in the fillets is probably fairly correct because such disturbing influences as present themselves seem to balance each other.

Another point of some importance has to be mentioned. In the forward crank fillets there are practically only bending stresses which act parallel to the axis of the shaft, alternating from a maximum tension to a compression of the same intensity and if the estimated alternating stress is, say, ± 5 tons per square inch, the range of alternating stress would be 10 tons. In the after crank-pin fillet the case is different, for when the after crank is at the top center both the torsion moment and the bending moment are at their maxima; at half stroke the torsion moment has died out, and the bending moment, being now at right angles to its first direction, produces practically no bending stresses in the crank-pin fillet. On reaching the lowest position the shearing stress due to torsion is again a maximum, and there is now a maximum longitudinal compression stress in the fillet, which is instantly changed into a tension stress when steam is admitted below the piston. On compounding these several stresses it will be found that the top position, the major right-angle resultant stress, is inclined, say, 10 degrees to 20 degrees, to one side of the axis of the shaft. It gradually dies out, and reappears as the compression stress inclined to the other side of the axis until suddenly it is changed into a tension stress and reoccupies its original direction. Thus the alternating tension and compression stresses do not act in the same direction and an estimated alternating stress of ± 5 tons per square inch in the after

crank is perhaps equivalent to a range of only 7 tons instead of 10 tons.

The result of the author's investigations may be roughly summarized by stating that if crankshafts are subjected to fewer alternating stresses their endurances can be estimated if the fatigue properties of the material are known; and seeing that modern materials for crankshafts have much higher fatigue enduring properties than was produced by wrought iron, modern crankshafts ought to be relatively much stronger than these. Some allowance should of course be made in the factor of safety for external influences, but it would generally suffice to adopt a factor of 1.5, taking the fatigue limit of the material as a standard.

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A COMPARISON BETWEEN THE RESULTS OF PROPELLER EXPERIMENTS IN AIR AND WATER.

By A. W. Johns.

Some years ago, in a paper read by the writer before this Institution, it was pointed out that, in the case of bodies moving through air at practicable velocities, the alteration of pressure is relatively so small that the compressibility of air—the quality in which it differs so materially from water—can be neglected. The coefficients of resistance for the same body moving in the two media—air and water—would thus be in the ratio of the densities. Theoretically this is not correct, but actually it appears to be sufficiently so for all practical purposes.

The rapid progress in aviation during recent years has been accompanied by the publication of a large amount of information resulting from the experimental work of the various aeronautical laboratories. So far as the writer is aware this information is not familiar to naval architects. It is with the object of emphasizing the importance of a knowledge and study of such results that this paper has been prepared. The performances of propellers are of practical importance, and in this paper the results of experiments with them in air have been compared with those obtained in water by Messrs. Froude and Taylor.

The propellers whose results are dealt with are five in number, all two-bladed. The diameter is in all cases 2 feet and the blade widths range from 6⅝ to 2⅝ inches. The shape of the portion furthest from the boss is elliptical, and that of the portion nearest the root is obtained by drawing tangents from the root of the blade to the ellipse.

The mean pitch ratio of all the propellers is .733. This is the nominal face pitch ratio. The published results, given by the Advisory Committee on Aeronautics, give the thrusts and efficiencies at various slips and various speeds of translation. The slips are calculated from the effective or vertical pitch, as used by Dr. Froude in his 1908 paper. A comparison of these results with the experimental data of Froude and Taylor shows a remarkable similarity in performance when the correction for different densities is taken into account.

It is found, however, that for constant slip, and on speed of translation base, that the thrust does not increase in all cases with the second power of the speed, and for narrow blades the power varies as at a greater power than the square of the speed, whilst for wide blades it varies at a smaller power than the squares. The efficiencies of the five propellers are given, and these are observed to increase with the speed, but as the slip is increased the speed at which maximum efficiency for the particular slip occurs gradually decreases. The increase of efficiency with speed is noteworthy, and may be of some practical importance if the same holds good for water. There is no reason, so far as one can see, why it should not. Mr. Taylor experimented at speeds of 3, 4, 5, 6 and 7 knots, but mentions no differences in the efficiencies as the speed varies.

Except in the matter of effective pitch, the aerial propeller results compare extremely well with those of water propellers. The thrusts are in agreement and can be obtained by dividing by the ratio of the densities of the two media. The efficiencies are in agreement at the lower speeds, and the results generally point to the fact that more experiments on propellers are required to clear up some important points. The increase of efficiency with speed, the departure of the thrust from the law of the square of the speed, the applicability of the law of similitude of Lord Raleigh, and the alteration of the slip at which maximum efficiency occurs as the speed is increased, are all of great practical importance. For this reason it is important and necessary that a careful study should be made of all results as they become available from the various aeronautical laboratories.

There are other points of interest to the naval architect in the results published by these laboratories. The best form of strut and the behavior of rudders are two which can be mentioned.

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THE EFFECT OF BEAM ON THE SPEED OF HYDROPLANES.

By Linton Hope.

This paper gives the results of the author's investigations to discover the fastest hull to take a 250 to 300-B.H.P. motor and only 1,100 pounds' weight. Three distinct types, namely, the multi-step, the single-step and the stepless hydroplanes or skimmers were investigated, and the result of these investigations lead the author to the conclusion that the single-step was much superior to the stepless, while the multi-step came somewhere in between the two. By a process of elimination and classification, the results given by the author were obtained. The various types were first tabulated according to ratios of $\frac{W}{P}$ and $\frac{V}{\sqrt{L}}$, and this eliminated the "wasters." They were then further arranged in order of the $\frac{B}{L}$ ratios, and it became apparent that, other things being equal, beamy boats had a decided advantage in speed, at any rate, when the ratios $\frac{V}{\sqrt{L}}$ and $\frac{L}{\sqrt{L}}$ were high and weather conditions favorable.

The author sought the advice of Dr. R. E. Froude, and submitted his formulae to him. Dr. Froude, although approving of the formula generally, strongly objected to the introduction of beam in any form, on the ground that only one dimensional factor was necessary or desirable, and that the proper proportion of beam for any given conditions would be determined by experience, while its introduction in the formula might lead to a wrong conception of its true value when designing future boats. Admitting this fully, there yet remains the fact that knowledge of the best proportion of beam to length is very slight, so, for analytical purposes only, the author ventured to go against Dr. Froude's advice, and used a constant varied by the ratio $\frac{B}{L}$, with surprising results.

The formula as corrected now reads:

$$B.H.P. = \frac{WV\left(\frac{V}{\sqrt{L}}\right)^{1.4}}{C}$$

for power, and—

$$V\ (kts) = \sqrt[2.4]{\frac{P.C.L.^{?}}{W}}$$

for speed. The coefficient C is obtained by multiplying the constant for each type by the ratio $\frac{B}{L}$; thus $C = 66{,}000 \frac{B}{L}$ for single-step hydroplanes, while for the stepless type it would be $55{,}000 \frac{B}{L}$, or thereabout. The type constant 66,000 or 55,000 would again vary according to the ratio $\frac{E.H.P.}{B.H.P.}$; but, as this appears to vary only from .47 to .48 in most of the examples, .475 was taken as a mean propulsive coefficient, and fixed constants of 66,000 and 55,000 for the two types.

Although the E.H.P. could not be obtained in many examples, some were obtained, notably the case of Sir John Thornycroft's *Miranda IV*, which has the highest efficiency of any example of a racing hydroplane which the author has come across. *Miranda's* ratio of E.H.P. to B.H.P. is .48, while the next highest (also a Thornycroft design) is .478, and most of the others appear to be about .475, except one which is as low as .47.

Taking .475 as the mean, it was necessary to increase the constant $60{,}000 \frac{B}{L}$ to $138{,}950 \frac{B}{L}$ to obtain the approximate E.H.P. necessary for towing experiments or for the floats of hydroplanes. For ascertaining the resistance only, the formula is simplified to

$$R = \frac{W \left(\frac{V}{\sqrt{L}}\right)^{1.4}}{K}$$

the value of the constant K being $425 \frac{B}{L}$ for the single-step and $355 \frac{B}{L}$ for stepless, while for the multi-step type it appears to be about $400 \frac{B}{L}$.

Table I gives tabulated data of the various types. All the thoroughly authenticated figures are printed in heavy type; those which have been obtained from reliable sources, but not fully authenticated, are in italics, and estimated figures are in ordinary type.

The *Flapper* was the first vessel to reach the speed of 40 knots, as the trials were run early in the year 1910. *Flapper*, which was designed, owned, and engined by Mr. M. E. Batting, was similar in type to Sir John Thornycroft's *Miranda III*, but much lighter in proportion, and her ratio of $\frac{W}{P}$ 13.44 pounds per horsepower, has only been beaten by the French competitor for the British International Trophy last year.

The author regrets he was unable to include the *Miranda III* in his table, as her best authenticated speed on the Seine is stated to be 51 knots. Although the following figures are believed to be fairly correct, the author was unable to verify them: Length, 7 m., or 23 feet; beam, 1.6 m., or 5.45 feet; weight, 3,360 pounds; power, 325 B.H.P.; and speed, 51 knots; ratio, $\frac{W}{P} = 10.34$ pounds per horsepower.

Returning to the table, after five columns of data there are three columns of the ratios—

$$\frac{B}{L}, \frac{W}{P}, \text{ and } \frac{V}{\sqrt{L}},$$

then in the next column the type constants—

$$C = 66{,}000 \frac{B}{L}, 61{,}800 \frac{B}{L}, \text{ and } 55{,}000 \frac{B}{L},$$

according to the classes A, B, C, and in the adjoining column the constant c obtained by the formula—

$$c = \frac{WV\left(\frac{V}{\sqrt{L}}\right)^{1.4}}{P}$$

The twelfth column gives the percentage of error between the last two columns, and the others give power and speed obtained by the formula with the percentage of error. It will be seen that, except in the case of two boats in Class B and two in Class D, the error is generally well under 1 per cent. Even in these cases the maximum error is only 1.69 per cent., while the average error of all the examples in Class A is 0.37 per cent., in Class B 0.85 per cent., and in Class C 0.25 per cent.

The power of the table is the brake horsepower of the motor minus 3½ per cent. for loss through the reduction gear.

No. 4 in Class B is also of interest, beyond the fact that she is the widest example obtainable, as she is the only one fitted with an aerial propeller similar to that of an ordinary aeroplane. Her performance is above the average, but it is impossible to say how much is due to hull design and how much to propeller efficiency. The difference, however, does not appear to be sufficient to warrant the claim so frequently put forward by aeronautical experts that aerial propellers are far more efficient than the ordinary marine type, unless the effect of beam has now become detrimental on account of the high ratio of $\frac{B}{L}$ 0.36. It is obvious that the beneficial effect of beam must have some limit, and the author expects that limit to be reached with a $\frac{B}{L}$ ratio of 0.3.

The author illustrates with a diagram the variation of the constant C for each type according to the ratio $\frac{B}{L}$. This diagram also shows some results of tank experiments made by Mr. G. S. Baker with five models of floats for hydro-aeroplanes. Unfortunately, these models are all very heavy in proportion to their length and power, their ratio of—

$$\frac{L}{\sqrt[3]{W}}$$

being 1.05 to 1.36, while the author believes that none of the real hydroplanes have ever obtained satisfactory results with this ratio below 1.7.

In the author's examples the average is 1.83, and runs up as high as 1.93, which amounts to a difference of 74 per cent. between the widest model and the average of the real boats of the same single-step type in Class A.

Not only were Mr. Baker's models much heavier in proportion than the real boats, but their power is also lower in proportion, and his mid-sections are more or less rectangular—a form long since discarded on racing hydroplanes. The proportionate speeds at which Mr. Baker's models were run were considerably below those which have given the best results in the racing hydroplanes, the average ratio of $\frac{V}{\sqrt{L}}$ being only 5.69 in the models, while it is 7.42 in Class A, the most successful type of the real boats.

The most interesting of Mr. Baker's experiments are the two models No. M. 43b and No. M 61, both of the single-step type Class A. The length and form of hull are practically the same in both, but No. M 61 is rather more than twice the beam of No. M 43b. Unfortunately, she is also double the weight, so her ratio of $\frac{L}{\sqrt[3]{W}}$ is far below that of any real

THE EFFECT OF BEAM ON THE SPEED OF HYDROPLANES.

TABLE OF DATA.

No.	L L.C.B.	B Beam.	W Weight.	H.P. B.H.P.	V Speed.	B/L	W/V³	V/√L	D Δ at 22 knots	V²/√W	Error per cent.	V²/W^⅓	Error per cent.	V/W^⅙	Error per cent.	No.	Remarks, &c.
	CLASS A.—STEPLESS STEP TYPE.																
1	ft. in. 30'00	ft. 6'00	lbs. 6,300	300·00	Knots. 42·00	·2000	30·00	7·276	15,101	13,643	Per cent. −0·70	268·05	Per cent. −0·28	Knots. 41·98	Per cent. +0·10	1	
2	30'00	6'00	6,303	180·00	32·00	·2143	37·73	6·414	14,103	14,003	−0·03	168·76	−0·03	35·10	+0·20	2	This boat had the engine and propeller of No. 1.
3	30'00	7'50	6,310	180·00	36·01	·2500	29·31	7·400	14,230	14,305	+0·10	169·21	+0·11	39·02	+0·06	3	
4	30'00	7'00	6,305	60·00	27·00	·2333	43·50	5·402	14,027	14,031	+0·34	69·07	+0·12	27·03	−0·12	4	Same boat as No. 3.
5	30'00	9'01	6,300	54·00	30·01	·3000	37·04	6·220	14,007	14,731	+0·51	84·20	+0·12	29·97	−0·12	5	Geared propeller.
6	60·00	9·00	11,700	120·00	30·20	·2000	19·33	7·020	14,012	14,026	−0·17	119·76	−0·17	30·20	+0·13	6	Speed only attained once.
7	15·00	3·00	672	18·00	41·00	·2027	13·44	10·109	14,093	14,000	+0·17	8·08	+0·16	40·96	+0·10	7	*Flapper* with plane.
8	30'00	6'00	6,650	53·00	28·50	·2000	40·15	5·569	13,333	13,375	+0·27	56·15	+0·27	28·47	+0·11	8	*Miranda IV.*
9	30·03	6·00	3,100	160·00	37·76	1994	20·14	6·676	13,333	13,362	+0·06	134·04	+0·05	38·01	−0·29	9	Ditto with small motor.
10	30'00	5'25	4,000	60·00	27·00	·1967	31·00	7·000	13,704	13,680	+0·43	68·39	+0·30	26·94	−0·16	10	
11	30'00	9'30	4,040	70·00	41·00	·3083	19·67	9·055	17,709	17,707	−0·03	77·91	−0·03	41·03	+0·04	11	
	$C = 61,800 \frac{B}{L}$																Note.—Formula and errors. All the results have been taken out by slide rule.
	CLASS B.—MULTI-STEP TYPE.																
1	30'04	9'03	1,925	60·00	30·05	·2000	31·19	5·474	12,300	12,300	−0·01	65·40	−0·21	29·79	+0·13	1	One hull with different powers and weights.
2	30'04	9'03	1,930	60·00	29·56	·2000	30·34	6·348	12,300	12,450	−1·05	65·24	−1·15	29·64	+0·03	2	
3	30'04	9'01	2,005	110·00	37·37	·2000	30·18	7·372	12,300	12,553	+1·15	115·07	+1·97	37·34	−0·01	3	
4	30'00	9'00	1,900	30·00	20·00	·2000	23·01	9·192	20,170	20,320	+0·75	30·10	+0·33	20·04	−0·15	4	Aerial propeller.
	$C = 18,000 \frac{B}{L}$																
	CLASS C.—STEPLESS TYPE.																
1	32'00	6'25	1,425	35·00	32·00	·2004	26·70	5·474	11,135	11,166	+0·30	35·10	+0·28	30·97	−0·12	1	
2	32'00	6'25	1,321	45·00	37·20	·2002	24·03	6·367	11,200	11,207	−0·28	44·69	−0·02	37·24	+0·13	2	} R.M.B.C., H.-P. Class.
3	32'00	6'50	1,370	45·00	37·00	·2143	25·47	6·330	11,787	11,316	+0·28	45·10	+0·28	37·04	−0·13	3	
4	32'00	5'40	676	30·00	32·00	·2027	13·44	9·811	12,405	12,405	−0·00	45·94	−0·12	32·02	+0·06	4	*Flapper* without plane.
	$C = 10,000 \frac{B}{L}$																
	CLASS D.—DISPLACEMENT TYPE.																
1	40'00	4'50	4,700	247·00	39·97	·1145	19·02	5·215	6,370	6,395	+0·39	262·07	+0·40	39·91	−0·13	1	*Dixie II.*
2	39·00	5·00	5,071	143·79	30·26	·1281	24·00	4·227	6,021	6,344	+1·13	147·07	+1·00	30·61	−0·14	2	*Aqua-Medallion.*
3	39'50	5'00	5,332	149·00	32·10	·1263	20·41	3·973	6,008	7,019	+1·00	149·05	+1·00	34·01	−0·03	3	*Medina II.*

skimmer and in addition she was not run at such a high speed as No. M 43b. If, however, she were run at the same speed of 22 knots, it may be assumed from the other examples that her power would decrease from 19.025 E.H.P. to about 15.0 E.H.P., or, say, 31.6 B.H.P., which would give her a constant by my formula of 18,608. As No. M 43b has a constant of 14,504 at 22 knots, doubling the beam has apparently increased the efficiency by about 21.4 per cent., in spite of the much lower ratio of $\frac{L}{\sqrt{W}}$ and a $\frac{B}{L}$ ratio which is probably well beyond the point of maximum efficiency.

This the author considers all in favor of his theory of a fixed $\frac{B}{L}$ constant, at any rate until $\frac{B}{L}$ reaches .3 or thereabouts; but it does not account for the extraordinary efficiency of Mr. Baker's toboggan-shaped stepless model No. M 98, which attained a speed of 24 knots with only 12.75 E.H.P. If we take her probable power at a speed of 22 knots to correspond with the two previous models, it appears to be about 14 E.H.P., or, say, 29.5 B.H.P., which gives her a constant of 9,893, whereas the formula for the C Class only gives her a constant of 7,895; so that even at 22 knots her constant is 25.6 per cent. better than that of any of the stepless racing hydroplanes, particulars of which are given in this paper.—"Shipbuilding and Shipping Record."

NAVAL VESSELS.

UNITED STATES NAVAL VESSELS UNDER CONSTRUCTION.

DEGREE OF COMPLETION.

No.	Vessel.	Building yard.	Engines.	No. shafts.	Speed, knots.	Percentage machinery completed 1915.		Percentage of completion May 1, 1915.	
						Apr. 1	May 1	Total.	On ship.
	BATTLESHIPS:								
36	*Nevada*	Fore River S. Co.	Curtis turbine	2	20.5	91.61	92.70	91.7	90.9
37	*Oklahoma*	New York S. Co.	Reciprocating	2	20.5	94.27	94.97	92.9	92.9
38	*Pennsylvania*	Newport News Co.	Cur. trb. grd. cr.	4	21	61.56	65.27	76 9	72.1
39	*Arizona*	Navy Yard, N. Y.	Pars. trb. grd. cr.	4	21	36.05	37.97	59.2	55.4
40	*California*	Navy Yard, N. Y.	.....	4	21	...	...	...	...
41	*Mississippi*	Newport News Co.	Cur. trb. grd. cr.	4	21	1.57	2.56	17.9	13.3
42	*Idaho*	New York S. Co.	Pars. trb. grd. cr.	4	21	4.18	8.60	25.2	10.0
	DESTROYERS:								
51	*O'Brien*	Wm. Cramp & Sons	Cramp trb. & rec.	2	29	97.77	97.77	97.3	97.3
52	*Nicholson*	Wm. Cramp & Sons	Cramp trb. & rec.	2	29	96.82	98.70	*	*
53	*Winslow*	Wm. Cramp & Sons	Cramp trb. & rec.	2	29	89 63	91.57	92.0	91.8
55	*Cushing*	Fore River S. Co.	Curtis trb. grd. cr.	2	29	95 16	95 97	92.0	91.9
56	*Ericsson*	New York S. Co.	Pars. trb. & rec.	2	29	95.88	96 27	96.3	96.3
57	*Tucker*	Fore River S. Co.	Curtis trb. grd. cr	2	29.5	57.84	59.72	61 7	57.7
58	*Conyngham*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	6..69	66.97	67.5	65.3
59	*Porter*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	54.96	63.05	66.3	63.9
60	*Wadsworth*	Bath Iron Works	Pars. trb. gearing.	2	30	91.82	93.90	91.2	91.2
61	*Jacob Jones*	New York S. Co.	Pars. trb. grd. cr.	2	29.5	73.30	78.69	68.0	67.5
62	*Wainwright*	New York S. Co.	Pars. trb. grd. cr.	2	29.5	75.14	78.44	66.9	66.4
63	.....	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	29.13	33.45	19.5	12.7
64	.....	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	29.13	33.45	18.6	9.8
65	.....	Bath Iron Works	Pars. trb. grd. cr.	2	30	4.66	12.06	10.5	3.9
66	.....	Bath Iron Works	Pars. trb. grd. cr.	2	30	4.66	12.06	10.5	3 9
67	.....	Wm. Cramps & Sons	Pars. trb. grd. cr.	2	29.5	...	12.12	7.3	1.5
68	.....	Navy Yard, Mare Isl'd	Pars. trb. grd. cr.	2	29.5	1.50	2.00	...	...
	FUEL SHIPS:								
13	*Kanawha*	Navy Yard, Mare Isl'd	Reciprocating	2	14	98.30	98.93	100.0	100.0
14	*Maumee*	Navy Yard, Mare Isl'd	Diesel	2	14	44.10	53.78	87.3	86.7
	SUBMARINES:								
31	*G-3*	Navy Yard, N. Y.	Diesel-Sulzer	2	14	92.00	92.00	88.6	88.4
40	*L-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	96 38	97.40	95.8	95.8
41	*L-2*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	96.38	97.40	97.0	97.0
42	*L-3*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	95.87	96.89	90.3	90.3
43	*L-4*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	95.61	95.99	90.0	90.0
44	*L-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	6.68	7.74	66 8	62.6
45	*L-6*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	6.46	6 48	59.0	53.6
46	*L-7*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	6.39	6.48	57.2	51.4
47	*M-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	89.86	90.83	73.3	68.1
48	*L-8*	Navy Yard, Portsmouth	Diesel-Sulzer	2	14	...	...	26 3	24.2
49	*L-9*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	80.92	85.77	66.0	58.1
50	*L-10*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	80.92	84.07	64.3	55.3
51	*L-11*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	66.91	71.16	52.7	42 0
52	.....	Elec. Boat Co., Quincy	Diesel-New Lond	2	20	...	....	...	...
53	*N-1*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	...	....	...	...
54	*N-2*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	...	....	...	...
55	*N-3*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	...	....	...	...
56	*N-4*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	...	....	14.5	11.4
57	*N-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	...	....	11.9	8 6
58	*N-6*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	...	....	11.2	7.9
59	*N-7*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	....	....	11.2	7.9
	SUBMARINE TENDER:								
2	*Bushnell*	Seattle Con. & D. D. Co.	Pars. trb. gearing	1	14	82.50	84.80	88.3	83.5
	DESTROYER TENDER:								
2	*Melville*	New York S. Co.	Pars. trb. gearing	1	15	87.00	89.15	88.0	87.9
	Transport	Navy Yard, Phila.	Reciprocating	2	14	3.61	4.41	13.4	6.3
	Supply ship	Navy Yard, Boston	Reciprocating	2	14	3.87	4.47	13.4	4.4
	PANAMA COLLIERS:								
	Achilles	Md. Steel Co.	Reciprocating	2	14	86.84	96.38	96.8	92.9
	Oil Barges 8 and 9	Navy Yard, Mare Isl'd	.....	...	...	88.70	99.98	...	...

* Delivered April 30, 1915.

BATTLESHIP *PENNSYLVANIA.*

The battleship *Pennsylvania*, launched from the yard of the Newport News Ship Building & Dry Dock Co., Newport News, Va., is the most powerful fighting machine yet added to the fleet of dreadnaughts of the United States navy.

The great advance made in naval design and construction is forcibly demonstrated when a comparison is made of the *Pennsylvania* and the dreadnaught *Delaware*, launched at the same shipyard Feb. 6, 1909. The *Delaware* has a displacement of 20,000 tons, and five turrets with two 12-inch guns each, a total of ten 12-inch turret guns; while the *Pennsylvania* has a displacement of 31,400 tons, and four turrets with three 14-inch guns each, a total of twelve 14-inch turret guns.

The *Pennsylvania* on Feb. 1, 1915, was 67.7 per cent. completed, and the contract date of completion is Feb. 28, 1916.

Some of the principal characteristics of the *Pennsylvania* are as follows: Length on load waterline, 600 feet; length over all, 608 feet; breadth, extreme, load waterline, 97 feet ½ inch; depth to weather decks, main and upper, respectively, 46 feet and 53 feet 9 inches; mean draught to bottom of keel at trial displacement, 28 feet 10 inches; number of decks, 5, with bridge deck and two platforms; number of compartments, about 400; mean trial displacement, 31,400 tons; speed, 21 knots.

Armament.—Main battery:—Twelve 14-inch, 45-caliber, breech-loading rifles. Four submerged torpedo tubes. Secondary battery:—Twenty-two 5-inch, 51-caliber, rapid-fire guns; four 3-pounder saluting guns; two 1-pounder guns for boats; two 3-inch field pieces, and two 0.30-caliber machine guns.

Complement.—Officers, 65; crew, 863; marine detachment, 74.

Fuel oil capacity, 694,830 gallons,—2,322 tons.

The plans and specifications for the *Pennsylvania*, authorized by act of Congress, approved Aug. 22, 1912, were completed and circular signed by the Assistant Secretary of the Navy Dec. 18, 1912, and issued to bidders upon request after December 20, 1912.

Contract for the *Pennsylvania* was signed with the Newport News Shipbuilding & Dry Dock Co., Newport News, Va., on Feb. 28, 1913, at a price of $7,260,000; to have the bidder's design of Curtis turbines installed, and to be completed within 36 months.

The total cost of completed ship is approximately $13,000,000. The keel was laid Oct. 27, 1913.—"The Marine Review."

U. S. S. *WADSWORTH.*

Destroyer No. 60, the *Wadsworth*, was successfully launched at the Bath Iron Works, May 1, 1915.

The *Wadsworth* is one of a class numbered from 57 to 62, contracts for which were let by the Navy Department in the latter part of 1913. The contract price of the vessel is $884,000 and the principal dimensions are as follows: Length, 315 feet 3 inches; beam, 29 feet 10 inches; mean draught, 9 feet 3 inches; displacement on trial, 1,050 tons. The contract speed is 30 knots.

The peculiarity of this destroyer is that it is the first one in the U. S. Navy to be entirely driven through gearing. The main propelling turbines are of the Parsons reaction type and drive twin screws by means of helical pinions and gears. These propellers are designed for 450 revolutions per minute and to develop a shaft horsepower of 17,500.

This vessel has accommodations for 5 officers and 95 men. There are four 4-inch guns and four torpedo-tube mounts, each carrying twin torpedoes 21 inches in diameter.

U. S. S. *MAUMEE.*

The U. S. S. *Maumee,* launched at the Mare Island Navy Yard, March 3, 1915, is the first large Diesel propelled ship for the Navy. The *Maumee* is a submarine tender, of 5,000 indicated horsepower, twin screw, and designed for 14 knots.

PANAMA COLLIER *ULYSSES.*

The Panama Collier No. 1, *Ulysses,* successfully completed her preliminary trials on April 15, 1915, averaging 14.99 knots per hour on her five high runs, and 14.605 for the 24-hour run. The *Ulysses* was built by the Maryland Steel Company under contract dated April 9, 1914, to be completed in 16 months. Contract speed was 14 knots.

U. S. S. *TUCKER.*

Torpedo-boat destroyer No. 57, the U. S. S. *Tucker,* was successfully launched at the Fore River Ship Building Company on May 4, 1915. The *Tucker's* machinery consists of turbines in combination with cruising turbines and reduction gear. There are two shafts, with one ahead turbine, one backing turbine and one cruising turbine on each shaft.

U. S. S. *CALIFORNIA.*

On April 24, 1915, the award of the propelling machinery for battleship No. 40, the *California,* was made to the General Electric Company. The contract price is $441,000, and besides the main propelling machinery includes main air and circulating pumps, blowers for ventilating motors and generators, and exciters for the main generators. The main propelling machinery will consist of two turbo-generators of about 11,000 kw. each, running at 2,100 r.p.m., and four double squirrel-cage induction motors, one on each shaft, running at 175 r.p.m. The generators will be quarter-phase, two-pole, and have a maximum voltage of 3,000 volts. The main machinery will be manufactured and tested at the works of the General Electric Company and installed by the Navy Yard force at Brooklyn, New York.

FRANCE.

THE LATEST FRENCH DESTROYERS.

Among the latest additions to the French Navy are the four destroyers *Aventurier, Opiniatre, Temeraire* and *Intrepide,* which are the largest vessels of their type now under the French flag. All of these destroyers were built and engined by the "Britany yard" at Nantes. Their main particulars are: Length overall, 283 feet; beam, 28.32 feet; depth, 17 feet; draught, at the stern, 10 feet; displacement, 1,100 tons.

The type of hull is the same as adopted in the previous destroyers *Voltigeur, Fourche,* etc., which have been illustrated and described in previous issues of this journal. This type of design, however, has proved very efficient, allowing the propellers to give their maximum efficiency even in bad weather. Their seaworthiness has been amply proved in actual service during the winter months.

The engines consist of two Rateau turbines, each driving a propeller 7

feet 3 inches diameter and 6 feet 9 inches pitch, working at 650 revolutions per minute. The total horsepower developed is about 20,000, and it is reported that speeds of 32 knots were obtained on the measured-mile trials. Steam is supplied by four water-tube boilers, using coal as fuel, and a fifth fired with oil fuel. The capacity of the coal bunkers is 12,362 cubic feet and of the oil tanks 3,179 cubic feet. The destroyers have an acting radius of about 3,000 miles.

The armament consists of four 4-inch rapid-fire guns, all mounted on the centerline of the vessel, and four 18-inch torpedo tubes.—"International Marine Engineering."

FRENCH BATTLESHIPS *BRETAGNE* AND *PROVENCE*.

The battleships *Bretagne* and *Provence*, laid down on May 1, 1912, and July 22, 1912, respectively, have recently carried out their official trials and now form a part of the Mediterranean fleet of the French navy, where they are the largest battleships in the fleet. The main particulars of these two battleships are: Length, overall, 545 feet; beam, 88 feet; draught, forward, 28 feet; draught, aft, 29 feet 3 inches; displacement, on normal draught, 23.546 tons; area of midship section, 2,752 square feet; brake horsepower, 28,000; contract speed, 20 knots.

On the *Bretagne* steam is supplied at a pressure of 256 pounds per square inch by twenty-four Niclausse boilers of an improved type, having a total heating surface of 65,000 square feet and a total grate area of 2,090 square feet. The funnels on this ship extend to a height of 66 feet above the load waterline. On the *Provence* steam is supplied at the same pressure by eighteen boilers of the Guyot du Temple type, having a total heating surface of 63,000 square feet and a total grate area of 1,500 square feet. This is the first time that boilers of the torpedo-boat type have been used on a ship of this size, and the performance of the *Provence* under severe service conditions will be followed with interest—especially so because some of the vessels of the *Normandie* class will also be fitted with this type of boilers.

In both vessels the main engines are Parsons turbines, arranged, according to the latest practice, on four shafts, without cruising turbines. The engines for the *Bretagne* were supplied by the Mediterranean Works and for the *Provence* by the Loire Works.

The electric plant consists of four generators, each of 200 kilowatts capacity. Forward and aft, near the turrets, there are also two other similar electric generators, making the total power of the generating sets 1,700 horsepower. All of the generators are driven by high-speed reciprocating engines.

Nearly all of the auxiliaries on the vessels are operated by electricity. Except in the engine and condensing rooms, all parts of the ship are ventilated by electrically-driven fans. The ammunition rooms are cooled by Westinghouse-Leblanc refrigerating apparatus.

The *Bretagne* and *Provence* are the first battleships of the French navy to carry an armament consisting of ten 13.5-inch guns all mounted on the centerline of the vessel.

GERMANY.

GERMAN SUBMARINES.

At the meeting of the French Society of Civil Engineers, held in Paris on March 26, M. Laubeuf gave the following data on German submarines: Information from the British Admiralty states that, at the commencement of 1914, Germany had 24 submarines ready and 14 in construction; of

TABLE I.—DIMENSIONS OF GERMAN SUBMARINES.

	U 1.	*U 2* to *U 8.*	*U 9* to *U 12.*	*U 13* to *U 20.*	*U 21* to *U 32.*	*U 33* to *U 38.*
Date of commencement	1903	1906–1907	1908	1909–1910	1911–1912	1913.
Displacement on the surface	185 tons	237 tons.	Slightly larger than the *U 2* boats.	450 tons.	650 tons.	675 tons.
Displacement when submerged	240 tons	300 tons.		550 tons.	800 tons.	835 tons.
Length	128 feet 3 ins.	141 feet 8 ins.		...	213 feet 3 ins.	...
Breadth	11 feet 10 ins.	12 feet 4 ins.		...	20 feet.	...
Draught	9 feet 2 ins.	9 feet 8 ins.		...	11 feet 10 ins.	...
Power of oil-fuel, surface, engines	400 E.H.P.	600 E.H.P.		1,200 E.H.P.	1,800 E.H.P.	2,500 E.H.P.
Power of electric under-water motors	240 E.H.P.	320 E.H.P.		600 E.H.P.	800 E.H.P.	...
Maximum speed on surface	11 knots	12 knots.		15 knots	16 knots	17 knots.
Maximum speed submerged	8 knots	8.5 knots		9 knots	10 knots	...
Radius of action on surface	.....	1,200 miles at 9 knots.		...	1,500 miles at 12 knots.	...
Radius of action submerged	.....	50 miles at 9 knots.		...	70 miles at 6 knots.	...
Armament	One torpedo tube. Three 17.7-in. torpedoes.	Two tubes. Four 17.7-in. torpedoes.	Two tubes. Four 17.7-in. torpedoes.	Two or three tubes. Four or six torpedoes. One 1.456-in. gun.	Four tubes. Eight 19.6-in. torpedoes. Two 3.464-in. guns.	... ...

these latter, eight were completed when war was declared, the six others—*U 33* to *U 38*—commenced in 1913, were certainly not finished when hostilities opened. On the other hand, Germany added to her own number of submarines five Austrian and one Norwegian boats, which were at the time almost completed at the Krupp Germania Yard, Kiel. The German Navy, therefore, had in all 38 submarines at the opening of hostilities. The first German submarine—*U 1*—was launched at the Germania Yard on August 30, 1905; this is largely an imitation of the French boats of the *Aigrette* type, commenced in 1902. The accompanying table gives some particulars of the German boats. As regards some of the later boats, particulars were given in the "Navy Annual." Length, 214 feet 1¼ inches; beam, 20 feet; displacement on surface, 750 metric tons; submerged, 900 tons; reserve of flotation at surface displacement, 56 per cent.; speed on surface, 20 knots, submerged, 10 knots; horsepower of oil motors, 4,000 brake horsepower, working on twin screws. It is thought that the five Austrian boats which Germany has kept have the same dimensions as the *U 33* to *U 38* boats. The Norwegian boat may be considered a replica of the *U 9* type. It is stated that Germany put in hand 20 new submarines at the end of 1914. In the summer of 1907 Germany had only one submarine in the service and seven in course of construction. The sum set apart for submarine construction in the 1907 budget amounted to £250,000. It increased rapidly, amounting to £350,000 in the budget for 1908; to £500,000 in that for 1909; to £750,000 in each of those for 1910, 1911 and 1912; to £1,000,000 in that for 1913; and to £950,000 in that for 1914. Germany undertook late in the day the construction of submarines, since her first boat, the *U 1*, launched in August, 1905, only entered the service in February, 1907. By this means the German Navy was able to benefit by the experience gained elsewhere. Her first twelve boats resembled the French type *Aigrette;* the eight following boats, of a larger type, resembled the larger Pluviose class. M. Laubeuf adds that, for equal displacement and age, the French submarines have better nautical qualities and a more powerful armament than the German ones. Notwithstanding the great activity displayed by Germany in submarine construction since 1907 and the importance of the sums voted for this work, she has not had time to build a very large number of submarines. Her program was to form by 1917 a flotilla of 72 boats; by the opening of hostilities she was able to put in service slightly more than half that number. Austria commenced the construction of submarines in 1907. When war was declared, she had six ready, *U 1* to *U 6*, of 300 tons, and four, on the Krupp designs, under construction at Pola. The five built by Krupp, above referred to, were to be numbered *U 7* to *U 11*. The four Pola boats have been completed, and are numbered *U 12* to *U 16*.—"Engineering."

JAPAN.

NAVAL OUTPUT OF JAPANESE SHIPYARDS IN 1914.

The most interesting vessel built in 1914 was the battleship *Fuso*, of 30,600 tons displacement, which was launched at the building dock at Kure in March, 1914. This vessel, which is one of the largest and most powerful battleships afloat, carries twelve 14-inch guns in six turrets, all arranged on the centerline of the ship. The vessel will go into commission during the latter part of next year.

In addition to the *Fuso* there are under construction three sister ships, namely, the *Yamashiro*, at the Yokosuka Navy Yard, the *Hiuga* at the Mitsu Bishi Dockyard at Nagasaki, and the *Ise* of the Kawasaki Dockyard of Kobe. The propelling machinery of these ships consists of Brown-Curtis turbines and Japanese Admiralty type water-tube boilers, with the

exception of the *Hiuga,* which is fitted with Parsons turbines. The battle cruiser *Hiyei* was completed last summer and is now engaged in actual service, while her two sister ships, the *Kirishima* and *Haruna,* are now undergoing speed trials at the Mitsu Bishi Dockyard and the Kawasaki Dockyard, respectively.

After the outbreak of the present war, a supplementary budget was passed in the special session of the Diet, held last September, authorizing the construction of ten destroyers of 600 tons displacement and 9,500 indicated horsepower, to be completed in about six months. Contracts for the construction of two of these vessels were placed with the Mitsu Bishi Dockyard, two with the Kawasaki Dockyard, and one each with the Osaka Iron Works and the Uraga Dock Company. The hulls, machinery and equipment of these vessels will be constructed entirely in Japan.—"International Marine Engineering."

PORTUGAL.

THE PORTUGUESE TORPEDO-BOAT DESTROYER *DOURO.*

The Portuguese Government, desiring to undertake the building of torpedo-boat destroyers at its arsenal at Lisbon, arranged a contract with Yarrow & Co., Limited, to supply it with complete designs of the hull so that the vessels could be constructed in Portugal, Yarrow & Co., Limited, supplying all the machinery and boilers and guaranteeing the full-speed and other trials.

The contract was for two vessels, of which the *Douro* was the first and the *Guadiana* the second. The first-named has recently completed all her trials and is now in the Portuguese service. So satisfactory have been the results that the Government has put in hand two identical vessels, the *Vouga* and the *Tamega,* for which the British firm is supplying the entire machinery equipment.

It is worthy of note that the success of the *Douro* is in a very great measure due to the high class of workmanship performed by the Portuguese workmen at Lisbon, together with the necessary supervision by the naval authorities. The hull of the vessel will stand the closest inspection, the plating and riveting being of the very highest class throughout.

The design of the *Douro* type followed that of the ten Brazilian destroyers constructed a short time ago by Yarrow & Co., Limited, but reciprocating machinery was discarded in favor of turbines. The length of the vessel is 240 feet; beam, 23 feet 6 inches; fully loaded displacement, 670 tons; and nominal shaft horsepower, 11,000. She is propelled by three screws driven by Parsons' turbines, constructed by Yarrow & Co., Limited, the high-pressure turbine being arranged on the center shaft, and the two low-pressure turbines with the reversing turbines on the wing shafts. There are three water-tube boilers of the latest Yarrow type designed for burning coal, one being double-ended and the other two single-ended. The only fuel used is coal. The question of oil fuel was considered, but, as its supply might not have been sufficiently reliable, coal was considered preferable.

The armament consists of one 4-inch quick-firing gun and two 12-pounder guns, as well as two 18-inch torpedo tubes, one of which is arranged for firing over the stern. Accommodation is arranged for five officers, and the total complement is seventy-two.—"The Engineer."

ASSOCIATION NOTES.

THE FOLLOWING MEMBERS AND ASSOCIATES have joined the Society since the publication of the last JOURNAL:

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Abbett, Henry J., Lieutenant, U. S. N.
Abbott, Henry L., Ensign, U. S. N.
Alexander, George A., Lieutenant, U. S. N.
Bankson, Lloyd, Naval Constructor, U. S. N.
Barleon, John L., Lieutenant, U. S. N.
Baughman, Courtlandt C., Lieutenant, U. S. N.
Bode, Howard D., Lieutenant, U. S. N.
Boucher, Creed H., Ensign, U. S. N.
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Cassard, Paul, Ensign, U. S. N.
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Durr, Ernest, Lieutenant, U. S. N.
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Foster, Murphy John, Ensign, U. S. N.

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Hewlett, George W., Ensign, U. S. N.
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Johnston, Frank L., Ensign, U. S. N.
Keisker, Herman E., Ensign, U. S. N.
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